PRODUCT ENGINEERING

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Things Are Not As Bad As They Seem

POVERTY is relative. It is measured by the degree to which the individual is unable to buy the available things that might satisfy his wants or desires. There is no definite level at which poverty begins. It is constantly going upwards, keeping in step with the highest "standards of living." The dire poverty of the share cropper today would have been considered affluence by the peasantry of France a century ago.

Engineers, translating the theories and facts discovered by scientists, are constantly developing new things that raise the "standard of living." Industry, finding better ways of doing things, constantly reduces the cost of producing them. Thus luxuries first become comforts and then necessities. Conversely, what was once considered a standard of good living is now classified as poverty.

In the past several decades new things have been made available at a speed never before experienced in the history of the world. There are now many more things to be had. The "standard of living" has gone up. This has accentuated the conditions of those who "have-not" and we look with horror upon present living conditions that are far better than the best that existed a century ago.

It is right that a civilized country should aim to have as many as possible share as much as possible in the national wealth of the country. Great progress in this direction has been made in the United States. Industry and surgery cooperated in the development of fever machines, X-ray machines, grain-of-wheat lamp bulbs for exploring inside the human body, and other new surgical apparatus. Through public clinics the benefits of this progress is made available to everyone. And to varying degrees, directly or indirectly, the benefits from every advancement in science and industry are reaching the great masses of the people.

But our scientific and industrial progress has been so rapid that when one considers all the things available, many are poor indeed.

Well meaning people, impatient to see an abundant life for everyone, are striving to find a formula by which the benefits of the progress of science and industry will be shared by all. This is a laudable aim. It is not new. The same motive resulted in the establishment of our free school system, public libraries, public clinics and medical foundations, agricultural bureaus, the Bureau of Standards and a host of similar public institutions. But the menace in the present crusade for greater prosperity for all is that the crusaders sometimes seek short cuts that defy laws of natural economics and also they often ignore psychological effects.

Most modern progress comes mainly from private industry. In research laboratories supported by corporations, often at a cost of millions of dollars per year, great minds develop new ideas, discover new principles and invent new things. Through the cooperation of industries, able management, economical design and labor saving machinery, the achievements of research are converted into serviceable products at ever decreasing costs and prices. That is progress. If allowed to continue, it is bound to spell greater prosperity for all.

Anything that robs industry of the incentive or capacity for continuing research will stop progress. Unfavorable legislation, exorbitant taxation, threats of public ownership and unnecessary government interference with business will do it. Merely the fear of these things will do it. Thus it is that ill-advised schemes to hasten wealth sharing may destroy the industrial researches and engineering activities that create the new sources of wealth. Industrial disintegration and a decline in learning would follow. But we'll never come to that.

CENTRIFUGAL CASTINGS

Qualities, Limitations and Possible Applications

Once used only for making iron pipes, centrifugal castings are now made in many shapes and of many materials. Cooperation of the American Foundrymen's Association and Ampco Metal, Inc., in the preparation of this article is acknowledged

FUNDAMENTALLY, centrifugal casting is a simple process. When molten metal is poured into a rapidly rotating mold, a number of things happen with the result that the casting produced has unusual properties and characteristics. As the centrifugal force throws the metal against the inside surfaces of the mold, thus forming the casting, the pressure created forces impurities in the form of oxides, gas and small particles of refractory to the inside surface of the casting because they are lighter in weight than the molten metal itself. These impurities are subsequently removed by a rough machining operation. Thus, by its very nature, the centrifugal process produces castings which are inherently sounder than sand castings.

Centrifugal castings may be produced in one of three ways. They may be made in a machine which revolves so that the metal is formed about a vertical axis which gives an outside shape impressed by the shape of the inside of the mold. Theoretically, the inside shape of the casting will be a paraboloid of revolution, but actually the vertical surface is to all intents and purposes a straight surface with little or no taper from top to bottom. This is particularly true of castings in the range of about twelve to twenty-four inches in length. Any departure from the normal shape is probably caused by freezing of the metal which prevents the downward pressure exerted by gravity. Generally the vertical casting method is confined to short objects such as gun shells and gear blanks.

Short casting can also be made in a machine in which the axis of rotation is horizontal. Such machines are used for casting long cylinders of various kinds, the axis of the casting machine coinciding with the axis of the object being cast. However, in the casting of piston rings, for example, the axis of the cylinder of the casting machine may be eccentric with respect to the axis of the ring being cast, in order to get the desired thin and heavy portions required. Other machines have been made to operate with an inclined axis, the resulting casting having an outside surface similar to the inside shape of the mold and an inside surface having a shape depending upon the inclination of the machine and speed of rotation. This method is used for long tubes where the inside shape does not matter or where it will be machined.

Most "spun" castings are of plain cylindrical shapes such as bushings, rings, dies, piston rings, pump liners and plain gear blanks. Special dies or molds have been developed for making flanged and tapered bushings, elevating nuts with outside diameters varying from two to six inches on the same castings, and for making web gears, with or without extended hub, and the web and face having different lengths or thicknesses.

Molds are made of many special materials and generally depend upon the practice of the contracting company making the castings. Special alloys are used to get certain desired cooling properties and in many cases baked sand liners are used.

With the metal being fed in from the center of the mold, solidification takes place from the outside of the casting progressively towards the center. Residual structure found in the center of all sand castings which solidify from both the outside and the inside surfaces is thereby avoided. In sand castings, impurities such as oxides, gases and slag are floated out of the casting and are frequently trapped by the freezing of the metal. The accompanying schematic diagram shows the effect of the two methods on casting structure.

Rotation, which may be as high as 2,000 to 3,500 r.p.m. produces a pressure which gives compacting and refining action resulting in a casting having hard and dense structure. Tests indicate that the physical properties are generally better with centrifugal castings than with sand castings. Tables I and II summarize the results of some such tests, Table II showing the improvement in properties with increase in size.

As a general rule, the higher the permissible rotating speed, the better and more dense the resultant casting. In alloys such as aluminum bronzes, tin bronzes and alloyed steels, wherein none

Centrifugally cast aluminum bronze screw-down nuts weighing 125 lb. each and with cast-in slots in the flange. To the right, gear rims weighing 12 lb. each. They demonstrate the possibility of casting large diameter rims with a small web

of the alloying elements have markedly different specific gravities, the limiting speed at which the machine can be run is the determining factor. When the alloying elements have widely different specific gravity the permissible speed of rotation is governed by the limiting speed at which these materials will not seriously segregate. This applies especially to metal mixtures such as the copper-lead bronzes in which no alloying takes place, the lead existing as a free metal. Since it is heavier than the copper it tends to separate and move to the outer surface when the molten metal is rotated at too high speed. Considerable care must be exercised in making castings of such "alloys" satisfactorily. However, in lead bronzes such as copper-tin-lead alloys in which alloying does take place between the lead and the tin as well as between the tin and copper. lead does not exist as a free metal and there is only a slight tendency toward segregation.

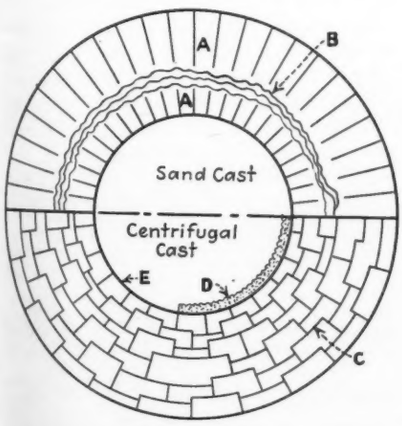
The structural density attainable in centrifugal castings makes the manufacture of cylinders, pipes and pressure vessels by this method particularly desirable. The reduction in porosity which results from the high pressures of rotation makes for less loss of liquid or gas under pressure. Tests have shown as much as twenty to one reduction in leakage by the use of centrifugal casting.

Centrifugal casting has been used to make blanks for transmission cluster gears and ring gears of steel and has been found to result in stronger, lighter and less expensive gears than those produced by conventional forging methods. In addition to the sound castings produced by centrifuging, free of flow holes and shrinks, any remaining crystal structure runs perpendicular to the line of action of the forces to which the gear teeth will be subjected when in use. Since the greatest grain compacting and metal purifying action is found at the outer diameter this method of casting is desirable both from the standpoint of strength and cleanliness.

The amount of waste material or the amount of metal which must be reworked is an important cost factor. In many instances the cast surfaces can be used without machining whereas the same part made as a sand casting would require some machining operation. Where machining is required in either kind of casting the centrifugal casting will require less because it can be made more accurately to dimensions originally. Thus the total cost of the centrifugal cast part when finished may be less than the total cost of the finished sand cast piece, although the pound price of the casting was more.

The decided element of purity and cleanliness of these castings is particularly advantageous when it is considered that money and time is often lost in the machining of castings before subsurface defects are discovered. Centrifugal castings are almost 100 per cent sound or acceptable from that standpoint so that there are no unnecessary machining costs.

Designers unfamiliar with the centrifugal casting process are likely to make the mistaken assumption that the die



Grain growth in typical sand cast section where chilling takes place from both inside and outside is shown at A *with the amorphous, weak, structure where the two grain growths meet, at* B. *Elongated radial grams have maximum strength in one direction only. Grain growth in a centrifugal casting where solidification takes place from the outside inward is shown at* C, *with the oxide inclusion forced to the inner surface at* D. *Small interlocking grains make a dense, strong structure. A minimum amount of material is removed to get a clean close to size surface, at* E, *of the rough bored casting as supplied to the customer*

TABLE I. Mechanical Tests on Pipes (Cast Iron)*

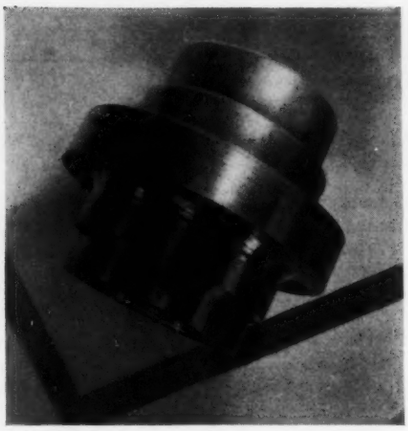
Diam. (in.)	Internal Pressure Test: Bursting Pressure	Internal Pressure Test: Calculated** Tensile (lb./sq. in.)	Transverse Breaking Test: Breaking*** Load (lb.)	Transverse Breaking Test: Deflection at Center (in.)	Transverse Breaking Test: Modulus of Rupture (lb.)
Centrifugal Cast					
6	2,800	28,500	31,400	1.13	45,600
8	2,500	29,800	50,400	0.80	41,000
10	2,250	30,400	77,300	0.82	41,400
12	2,000	30,200	103,000	0.64	34,050
Sand Cast					
6	2,050	15,600	21,000	0.97	27,200
8	2,200	19,850	33,600	0.62	21,600
10	2,050	21,400	76,100	0.64	26,600
12	1,550	18,000	100,800	0.55	25,600

* Fox and Wilson, Stanton-Lavand, Foundry Trade Journal, Jan. 21, 1926.
** Sand cast pipe was approximately 20 per cent thicker than centrifugal pipe.
*** Pipes 12 ft. long, load applied at two points, 4 ft. apart at equal distances from support points 10 ft. apart.

TABLE II. Influence of Size and Process on Bronze Gears*

Property	Method of Making: Sand Cast	Ring Chill	3-Sided Chill	Cent. Cast
Diam. 12 in., face 2 in., bore 8 in.				
Yield Point	16,000	19,000	26,000	37,000
Tensile Strength	30,000	33,000	38,000	44,000
Elongation, Per Cent	7	4	6	7
Hardness (10 mm. ball 1000 kg. load)	75	85	100	110
Izod (ft. lb.)	34	23	17	25
Compressibility, permanent set at 50,000 lb. per sq. in	6.8	4.5	2.9	2.6
Density	8.4	8.5	8.65	8.75
Diam. 18 in., face 3½ in., bore 12 in.				
Yield Point	14,000	15,000	23,000	31,000
Tensile Strength	26,000	30,000	34,000	41,000
Elongation, Per Cent	5	5	7	9
Hardness (10 mm. ball 1000 kg. load)	63	78	85	98
Izod (ft. lb.)	38	29	24	34
Compressibility, permanent set at 50,000 lb. per sq. in	8.2	8.3	4.2	3.5
Density	8.2	8.3	8.45	8.6

* Francis W. Rowe, Metal Progress, June 1933.



Pinion-gear blanks, centrifugally cast at a cost of only 6 cents per lb. greater than that of sand castings. To the right, bushings for controllable pitch propellers—as cast, as shipped and as finished. Dependable sound castings are the first consideration

cost and the set-up charges involved make this process applicable only to quantity production. Under the modern methods used in the manufacture of centrifugal castings the die or mold cost is usually not much different from the cost of good sand casting pattern equipment. Set-up charges apply only for lots of fewer than ten pieces where the weight involved is not more than 50 lb. per casting. As an off-set to whatever die cost and set-up charges are involved, the metal yield is much higher for centrifugal castings than for sand castings so that the resulting saving in metal cost frequently more than offsets the die and mold expenses. This is particularly true for parts wherein the wall thickness is relatively great; the thicker the wall the greater will be the per cent of metal yield. Of course, this is also true in sand castings but to a lesser degree.

There are certain limitations in size of casting because of the fact that die temperatures must be well controlled in order to secure a quality casting. It is difficult to control this temperature if the castings are too small. Where there is any doubt as to design of small parts the experience of the centrifugal casting manufacturer will indicate the possibilities of using the method.

The stresses and forces set-up in a centrifugal casting machine requires a sturdy machine, a good foundation and rather a good balance for best performance. For that reason centrifugal castings have been confined to those which are in balance diametrically and usually having regular outer and inner forms.

This limitation on shape is perhaps the greatest limitation on the general adoption of centrifugal castings. As a better quality of casting is insured with the higher speed it is desirable that the weight of the casting and of the machine be well balanced about the machine's axis. To keep the unbalancing loads at a minimum it is required that the castings be diametrically balanced with a cored hole and a uniform inside diameter. No taper on the inside surface can be expected and irregular shapes on the inside surfaces such as bosses on the inside of a fly-wheel rim can be put in only with special molds. Attempts have been made to overcome these faults by making molds in which balancing weights are used on the opposite side to overcome the effect of an unbalanced casting.

Some special cases will further illustrate the possibilities of this method. Some years ago Campbell, Wyant & Cannon Foundry Company were experimenting with brake drums for automobiles. It was desired to retain the lightness and appearance of the outer pressed steel drum plus the braking advantages of an inner cast iron lining. Good cooling properties were essential and a uniform load between the lining and the band gave the best results. After experimentation it was found that the lining could best be cast and fused into the steel housing by a centrifugal process.

Hardness on the rim of car wheels has been secured by using a centrifugal casting procedure. Manganese powder is metered into the mold in the early stages of pouring. As the first metal going into the revolving mold forms the rim the manganese fuses with it giving good hardness and wear properties where they are needed.

High-chromium steels were first made into seamless tubes by centrifugal casting because of the difficulties in extruding them. The method also permitted the manufacture of larger tubes of this material than would have been possible by other methods. A tube 13 in. in diameter with $\frac{5}{8}$ in. wall and 12 ft. long has been cast centrifugally.

In general, the centrifugal casting method has possibilities in the field where lack of ductility of a metal makes it difficult to form by rolling, forging or some other method. It is also of great service in the production of parts out of viscous materials. Such materials are more prone to entrap oxides than, for example, steel, their lack of fluidity preventing them from easily releasing gases and other impurities. Copper-lead alloys fall in this class of material. Aluminum bronzes are viscous and require particular care in founding to get good results with sand casting. Centrifugal casting also has possibilities for thin castings where the flow will be assisted by centrifugal force which minimizes necessity for super-heating with attendant possibilities of unsound castings due to increased gas absorption.

Summing up the advantages of centrifugal casting there are such structural benefits as dense, non-porous structure, additional strength of material, the possibility of producing special castings where certain qualities are desired, the saving in metal which the more accurate method of centrifugal casting provides, consequent reduction in machining time, less scrap metal in the molding process, and no waste of machining time such as occurs on castings which have sub-surface defects.

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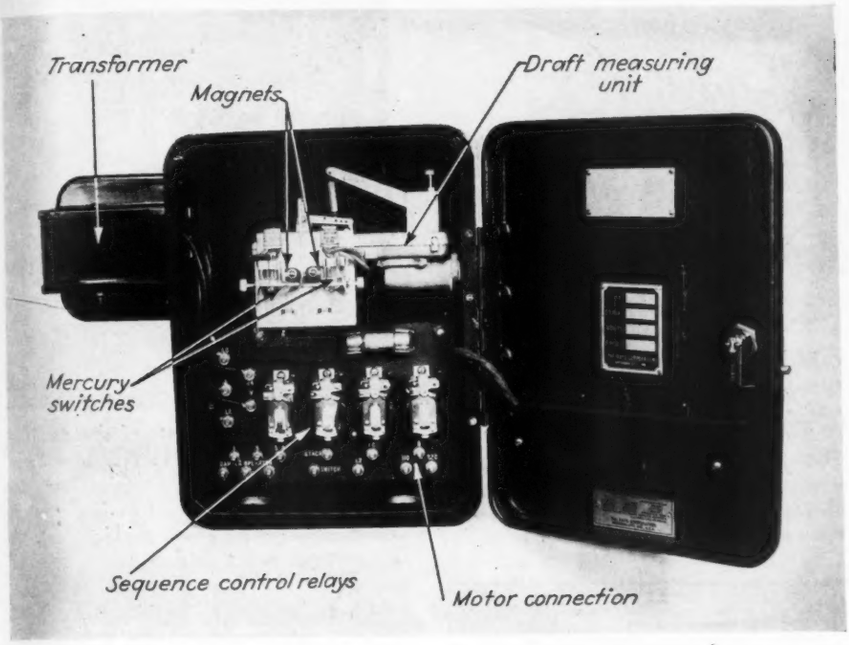
Current Quotations

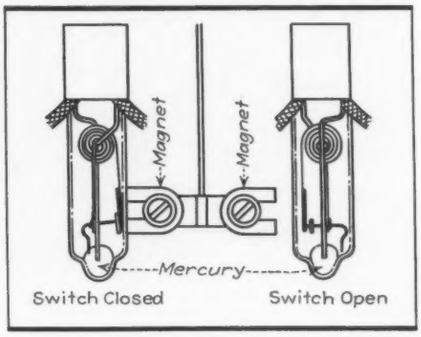
"Though he deals only with the facts of science and industry, it is the daily task of the designer of machinery and other products to see visions and dream dreams. Thus he bridges the gap between scientific progress and commercial fulfillment, ever developing new marvels and converting the luxuries of yesterday into the commonplace comforts and necessities of the day."—— HARVEY N. DAVIS, *president, Stevens Institute of Technology and president of the A.S.M.E., in an address before the Power Transmission Council.*

"Nineteenth century merchandise will not meet twentieth century specifications. New developments in scores of industries inevitably reflect demands for unusual properties or characteristics in capital goods."—— RUFUS E. ZIMMERMAN, *research vice-president, United States Steel Corporation, in an address at a meeting of the Iron and Steel Institute.*

Modern Designs

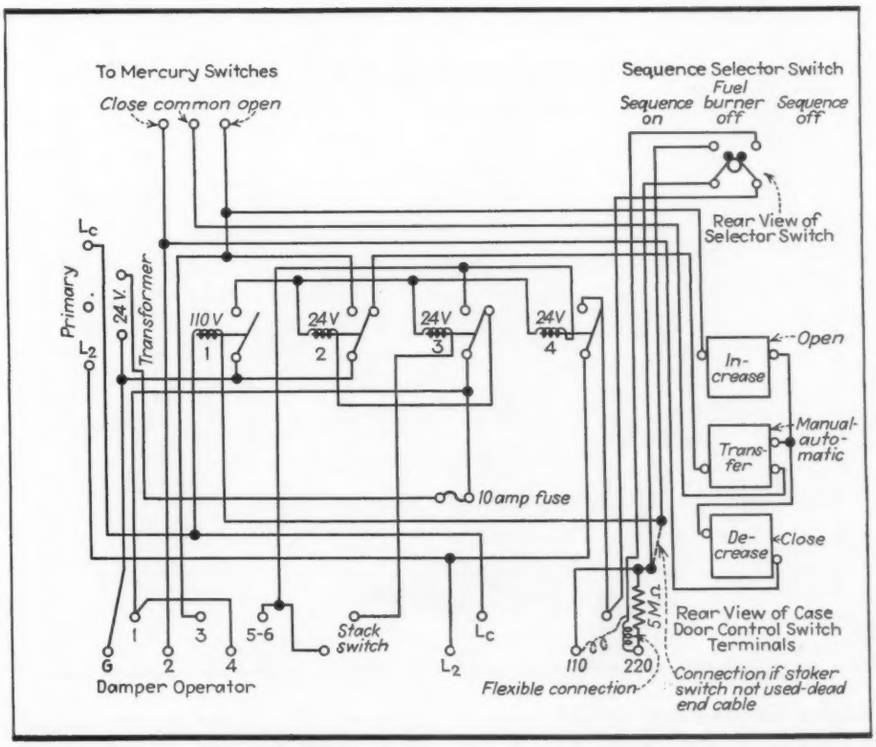
Centralized Draft Control





Pressure variation on the diaphragm changes the position of the magnet arm, swinging the magnet to the right or to the left. As the magnet approaches the switch it attracts an eye plate through the glass tube, which draws the wire into a globule of mercury, closing the electrical circuit. This starts a motor which moves the damper and corrects the condition that caused the pressure variation on the membrane.

Sequence control developed by the Hays Corporation of Michigan City, Ind., prevents "blow-back" that often occurs at the ignition period in the "on and off" type of oil and gas furnaces and stokers. The control is electrically operated and is timed so as to open the up-take damper wide an instant before the burner or stoker comes on. No pressure is built up in the combustion chamber and it is claimed the ignition is accomplished without the usual explosion. Immediately following the establishment of combustion, the damper is closed to a position required to maintain good combustion, and the regular draft control maintains the proper draft. Diaphragm, held in a die cast aluminum housing, has bearings of hardened beryllium copper. Slight variation from predetermined pressure causes an immediate corrective adjustment. Sequence control responds to pressure, temperature, or main control switch, closing relays 1 and 2 which open the damper to its full position. Firing equipment then starts and damper is then controlled by the draft control unit. Three toggle switches are provided for manual operation of the control. All electrical parts, except the transformer, are housed in a dust-tight cast-iron case with hinged cover.



Control unit consists of a set of fully inclosed, magnetically operated fixed mercury switches, a dry type draft measuring unit with a slack-leather diaphragm, transformer, relays, and the necessary connections.

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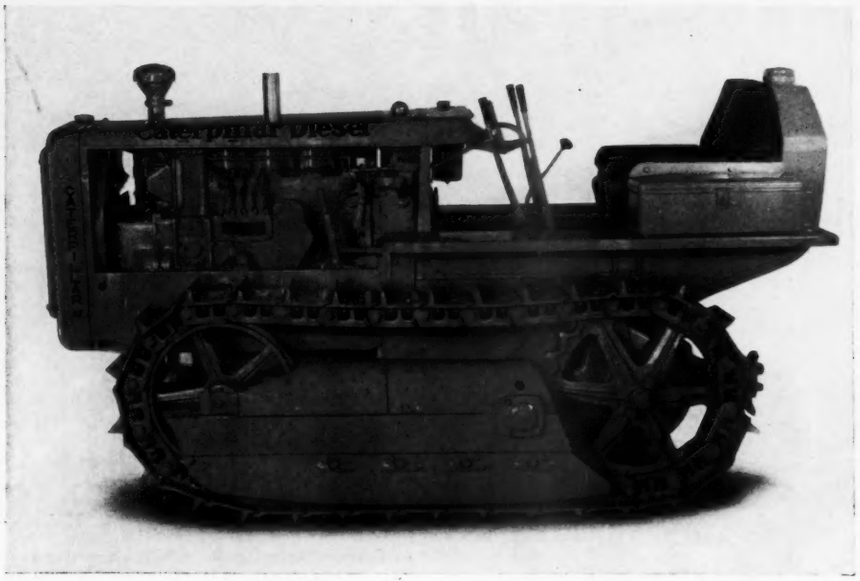
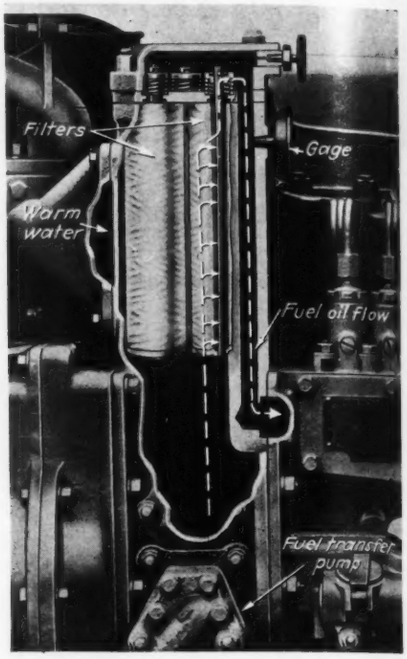
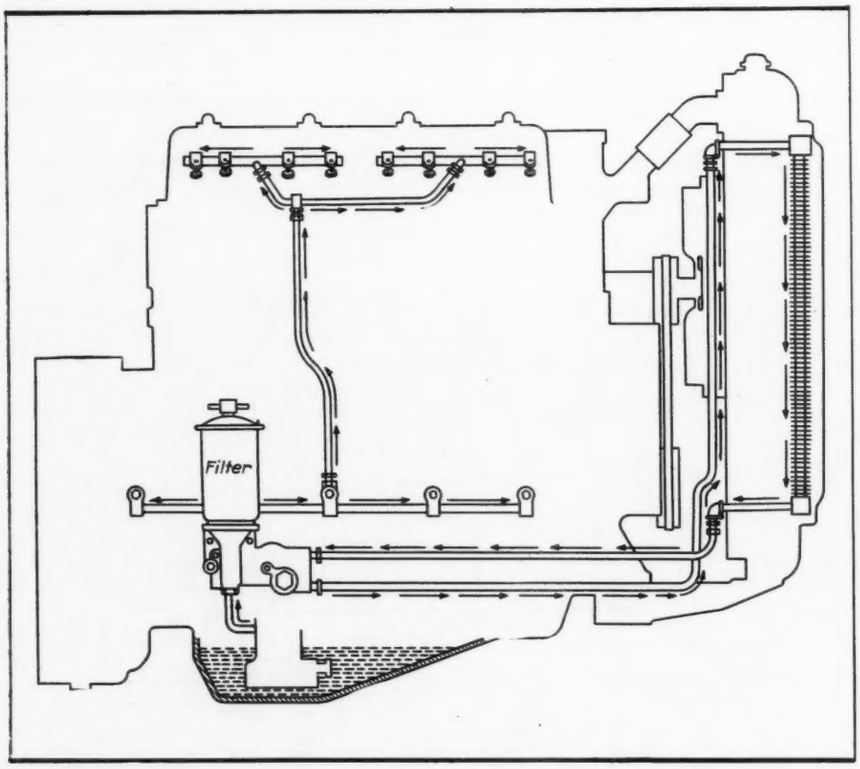
Modern Designs—Tractor Radiator Air-Cools Crankcase Oil



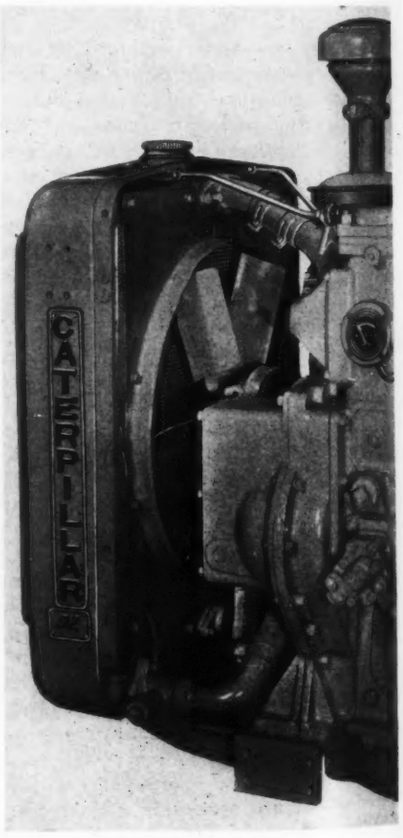
Smallest diesel track-type tractor, built by Caterpillar Tractor Company, incorporates larger tractors' features such as five speed transmission four forward speeds ranging from 1.7 to 5.1 miles per hour, and reverse speed of 2.1 miles per hour. Diesel is warmed up and started by an independent two-cylinder, horizontal opposed four-cycle gasoline engine of 10 hp. at 3200 r.p.m. which drives through a multiple disk clutch and helical gears to the flywheel. The pinion automatically disengages when the diesel starts. The fuel tank also serves as a seat for the driver. Aluminum alloy pistons are used.

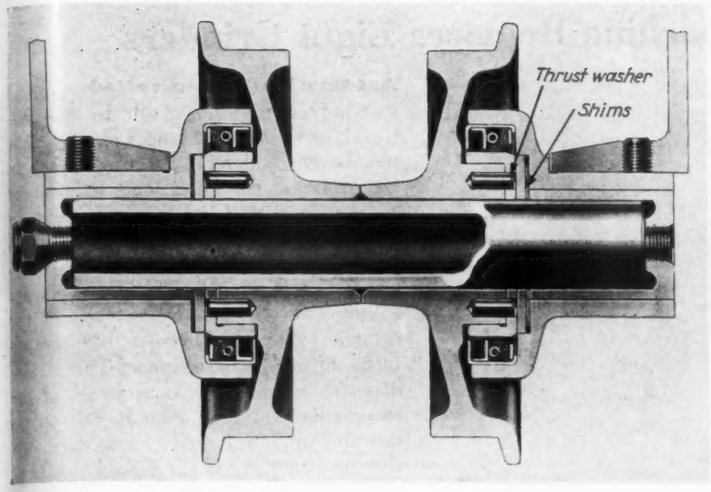


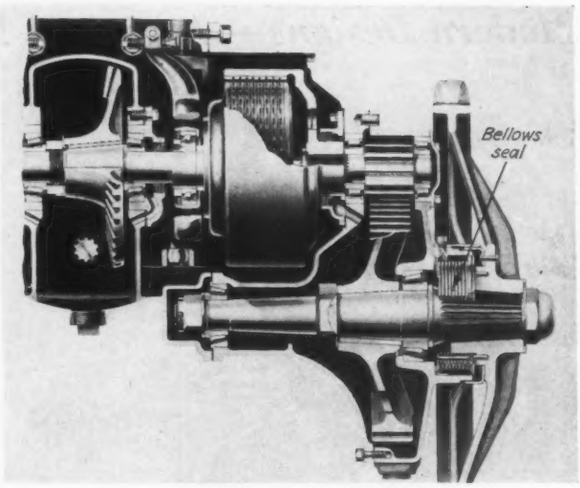
Fuel oil filter consists of three cotton string elements made in cartridge form for easy replacement. Pressure line connects filter to gage which indicates when filter is dirty and needs replacement. A connection leading from the hot water system carries warm water to the filter, for warming fuel oil under extreme cold conditions.



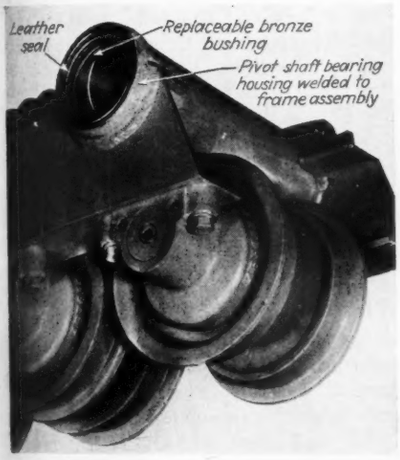
Engine oil cooling, claimed to give a material increase in bearing life, is accomplished by circulating the hot oil through the twin radiator, then to the oil filter and distributing manifold under pressure. The oil radiator of flattened finned tubes is placed in front of the water radiator. A thermostatic valve prevents oil from passing through the radiator when cold.

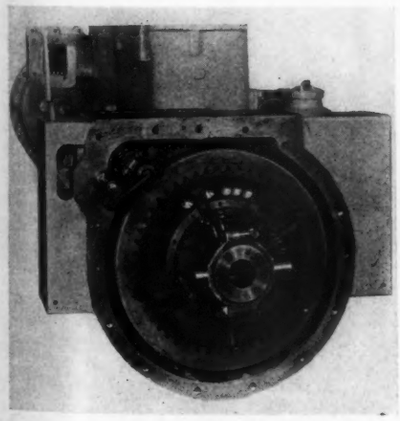




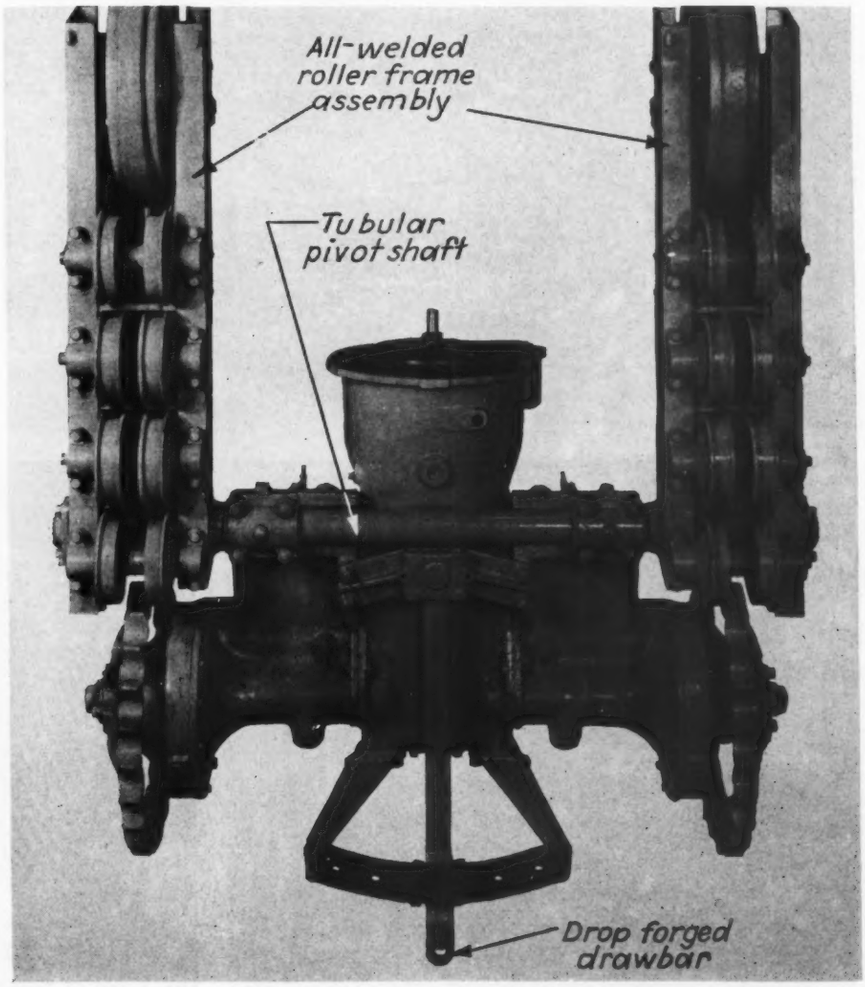


Roller rim and hub are forged steel, heat-treated, and shrunk onto a hardened steel shaft. Bronze bearings pressed into the end collars hold rollers rigid. The inside of the live shaft acts as a lubricant reservoir since lubricant can be forced into it by a pressure gun through a buttonhead pressure fitting located in the bronze bearings. Bearings are protected by twin seals; the outer seal keeps out dirt and water. Back of this is a spring reinforced leather seal, which stops water or dirt that gets by the outer seal and keeps in the lubricant that seeps past the thrust washer. End thrust is taken by two hardened steel washers. Bearings supporting pivot shaft are replaceable bronze bushings protected by leather seals. Final drive is kept clean by a bellows seal which allows end play, but seals out dirt.



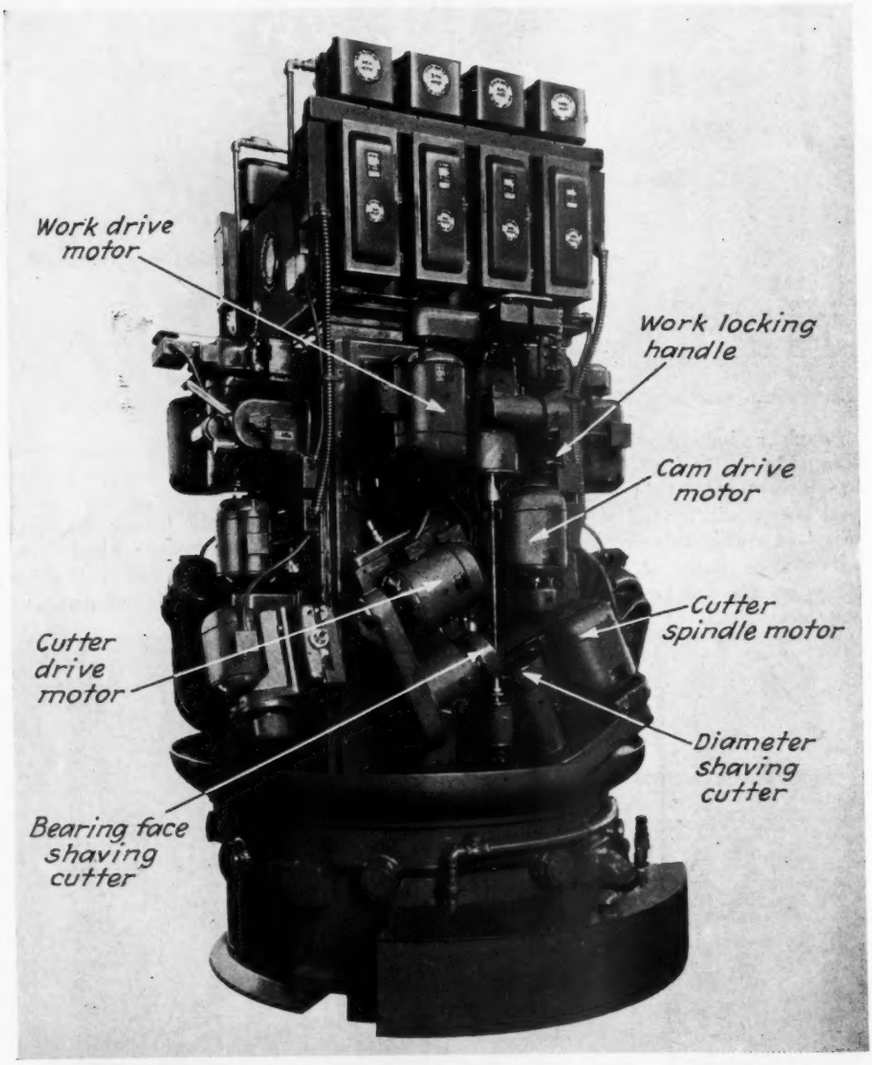


Dry clutch is hand-operated on this tractor. When clutch is engaged, it automatically locks the gear shift lever in place for working over rough ground.



Track roller frame assembly is of all welded construction. Track recoil spring is located on top of the track roller frame with a track-adjusting nut located back of the idler. No lubrication is required in the entire recoil spring mechanism and a single adjustment maintains equal tension on each track rail.

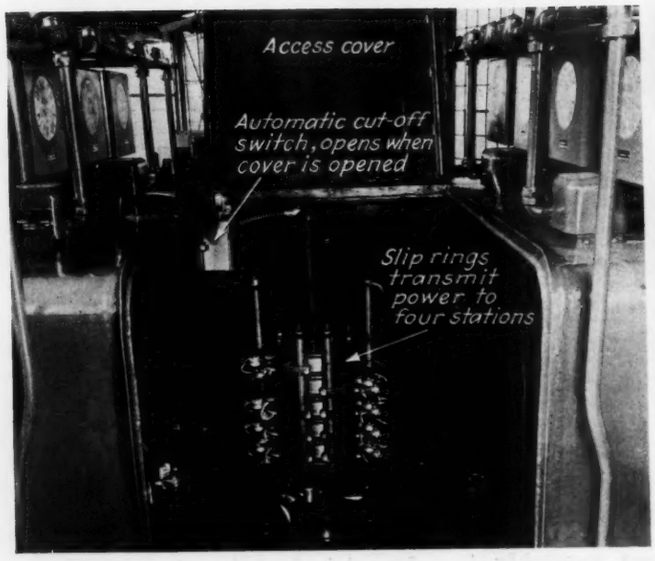
Modern Designs—Automatic Machine Replaces Eight Grinders



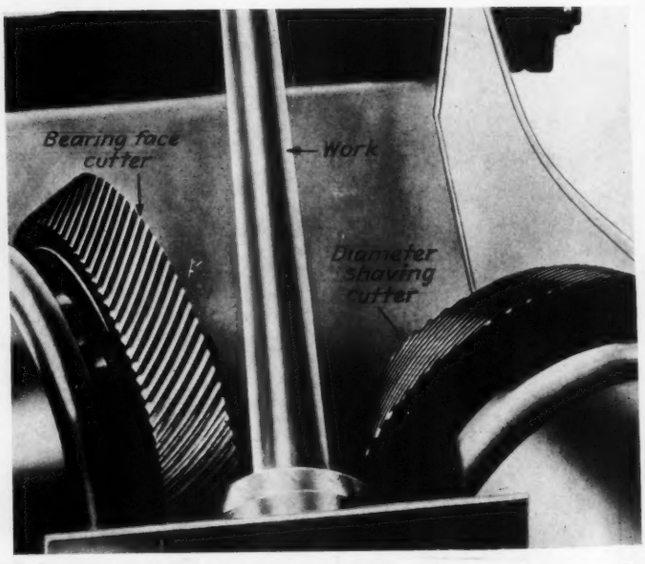
Automatically controlled, vertical station-type machine, built by the National Broach & Machine Co., with four panels disposed on a rotating rectangular column, machines one axle every 7 seconds with finish comparable to a ground finish and accuracy maintained to within 0.001 in. All operating elements except the electrical controls are mounted on the outside surface of the column to provide accessibility for adjustment and maintenance. Three spindles are mounted on each panel of the rectangular rotating column, one to rotate the work unit and two to drive the shaving cutters of Na-Loy steel. A limit switch interlocked with the load clamping handle prevents operation if the panel is not loaded.

Each column panel carries its own forced feed lubrication system. Coolant supply is shut off in loading position automatically by a closed portion in the oil circuit. All slides are hardened and ground and all spindles adjustable for wear. V-belt drives and anti-friction bearings are used throughout the machine.

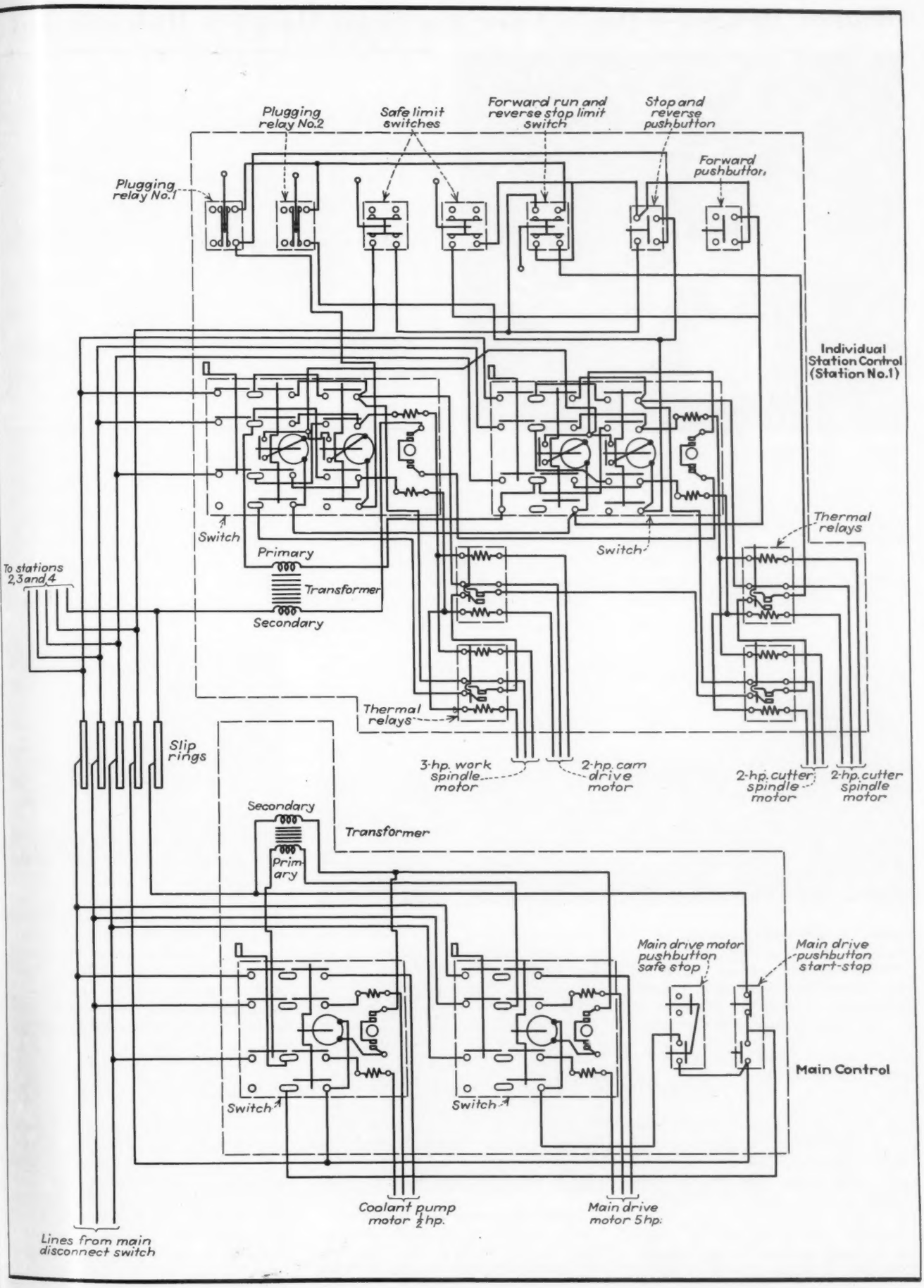
Wiring diagram, on facing page, shows how eighteen standard motors are controlled by plugging relays which prevent coasting when the automatic control mechanism cuts off the current.



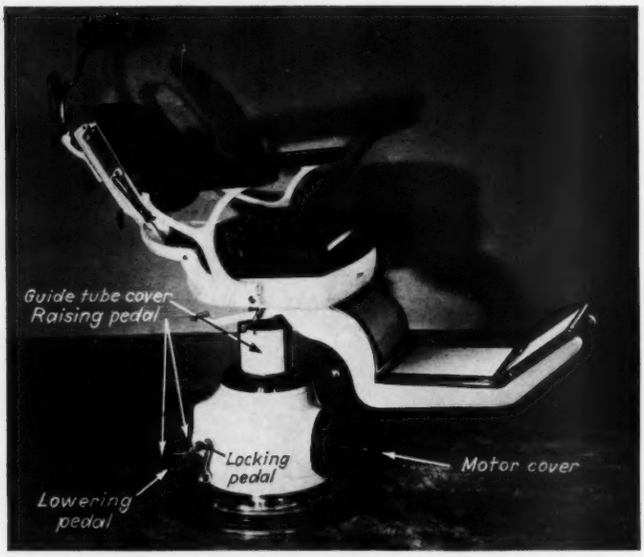
Electrical relay system and slip-rings are housed in a covered compartment at the top of the column. An automatic switch, shown in the upper left corner of the photo, cuts off all current to the machine when the cover is opened.

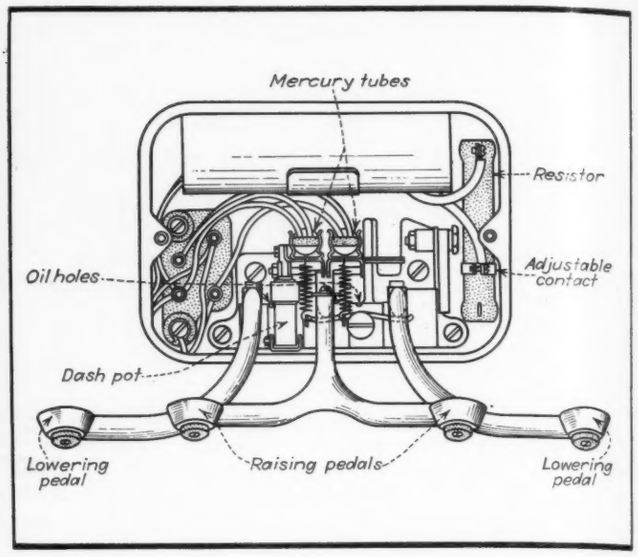


Operating simultaneously on the rotating unit, one cutter shaves hub diameter, and the second shaves the flat bearing face. The shaving cutters are moved into position against the rotating work by cam operation.

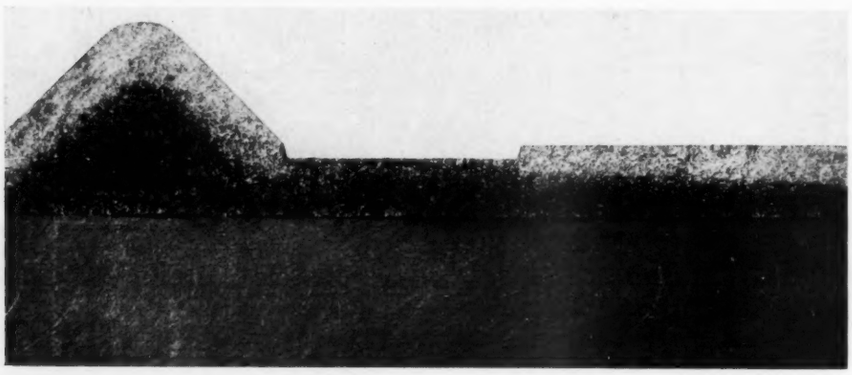

Plugging relay No.2
Safe limit switches
Forward run and reverse stop limit switch
Stop and reverse pushbutton
Forward pushbutton
Plugging relay No.1
Individual Station Control (Station No.1)
Switch
Primary
Transformer
Secondary
To stations 2,3 and 4
Switch
Thermal relays
Thermal relays
Slip rings
3-hp. work spindle motor
2-hp. cam drive motor
2-hp. cutter spindle motor
2-hp. cutter spindle motor
Secondary
Transformer
Primary
Main drive motor pushbutton safe stop
Main drive pushbutton start-stop
Main Control
Switch
Switch
Coolant pump motor ½ hp.
Main drive motor 5 hp.
Lines from main disconnect switch

Modern Designs— Dental Chair Has Motor Operated Hydraulic Lift

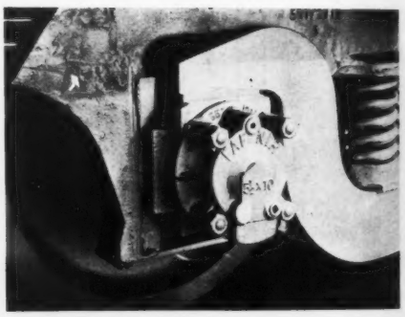




Electrically operated hydraulic lifting mechanism in the Ritter Dental Chair has an a.c. capacitor type or a d.c. type motor, depending upon the power supply available, driving a Tuthill gear pump. Two sets of operating pedals are provided, with the two middle pedals starting the motor and elevating the chair, and the outside pedals lowering the chair, without the aid of the motor, by releasing oil from the lifting tube. Smooth operation, when elevating the chair, is insured by employing two mercury tube switches in conjunction with a resistor. Depressing the raising pedal actuates both mercury tubes, the action of the left-hand tube, however, being delayed by the dashpot. When the right-hand mercury tube closes the electric circuit, the current passes through the resistor before reaching the motor. This action is followed by the closing of the second mercury switch which disconnects the resistor from the circuit and effects the gradual acceleration of the motor to full speed. Resistor may be readjusted for extreme voltage variation.



Flame hardening, applied to a Monarch "V" "Flatway" bedway, is shown in this cross section. The depth of the hardened portion is $\frac{1}{8}$ in. or more, minimizing the possibility of injury to the way surface from chuck or work falling on the bed. It is claimed that flame hardening does not weaken the bed section; also sections of the bed need not be planed off to receive hardened inserted way strips, bolted or dove-tailed to the main bed section.



Combination ball and roller journal box, designed by Fafnir Bearings, Inc., uses high capacity thrust ball bearings to take thrust loads caused by high-speed operation on curves. Radial load is carried on hardened solid rolls. Other design features include hardened alloy housings, reversible for double wear, and heavy rubber pad noise insulation.

Random Jottings About New Developments

Stapling wire known as Colorstitch is now manufactured in practically all colors. The wire is first galvanized and then colored on one side. It is claimed that the finish will not chip or peel.

In order to keep its station and round houses clear of smoke, the Chicago, Burlington & Quincy Railroad employs a testing device which draws samples of air through monel tubes fitted with an electric eye which records the smoke density and cinder content.

Electrically operated hoist which controls huge spillway gates on the principal dam on the lower Colorado River Reclamation Project near Austin, Texas, is constructed of mild rolled steel shapes and plates are welded together. I-beams are used for the longitudinal frame members and other structural parts are steel plates cut to shape. The hoist was fabricated by the Leitelt Iron Works, and welded with shielded arc equipment supplied by the Lincoln Electric Company.

WORKING STRESSES

In Members Under Fluctuating Loads

JOSEPH MARIN, *RUTGERS UNIVERSITY*

Based on a theory that failure is a function of the shear energy a procedure is developed for determining allowable values of the stress components in terms of a given working stress

IN SHAFTS, axles and many other machine members the resisting strength of the material is reduced in time because of fluctuating loads. This change is an important consideration when calculating working stresses in parts subjected to fluctuating loads.

Usually fluctuating tensile stress can be considered, as shown in Fig. 1, as a mean stress S_m and a variable stress $\pm S_r$ combined in such a way that the stress value is S at a time t and having a sinusoidal variation with time. This condition usually occurs in both the tests and applications. Different combinations

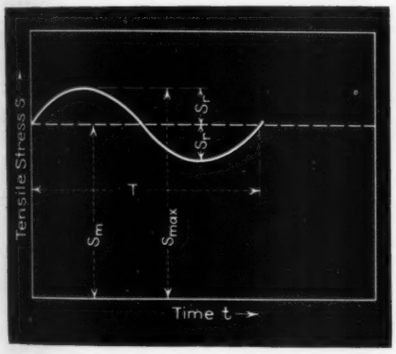


Fig. 1—Fluctuating tensile stress considered as a combination of a mean stress and an alternating stress which has sinusoidal variation

of the mean stress S_m and the variable stress S_r can be represented by coordinates as shown in Fig. 2, in which the vertical axis represents the maximum stress S_{max} and the horizontal axis the mean stress S_m. When S_r equals 0 the stress is entirely static and for ductile materials failure will be defined when the stress S_m equals the yield of the material S_{yp} which is represented by point C in Fig. 2. When S_m equals 0 the stress varies from $+ S_r$ to $- S_r$ and there occurs complete reversal of stress so that the material fails at a stress S_r equal to the endurance limit for complete reversal of stress S_e which point A in Fig. 2 represents.

Between these limiting conditions the maximum stress at failure has a value different from either S_{yp} or S_e. It will be assumed that the plot of the values of S_{max} and S_m for intermediate conditions is a straight line joining points A and C, Fig. 2. This assumption is justified since it is on the safe side of the test results studied; see "Working Stresses," C. R. Soderberg, Journal of Applied Mechanics, *Trans.* A.S.M.E., Vol. 57, p. A-106, and also "Working Stresses for Members Subjected to Fluctuating Loads," J. Marin, Journal of Applied Mechanics, *Trans.* A.S.M.E., June, 1937, p. A-55. The equation of the line AC is

$$S_{max} = (1 - m)\, S_m + S_e \qquad (1)$$

where

$$m = S_e/S_{yp}$$

For a given value of S_m the allowable value of S_{max} is represented by the line DE in Fig. 2. The equation of this line is

$$[S_{max}]_w = (1 - m)\, S_m + S_e/n \qquad (2)$$

where $[S_{max}]_w$ equals the allowable or working value of S_{max} for simple stresses and n equals the factor of safety.

Before defining working stresses in conjunction with combined stresses, it is first necessary to formulate a theory

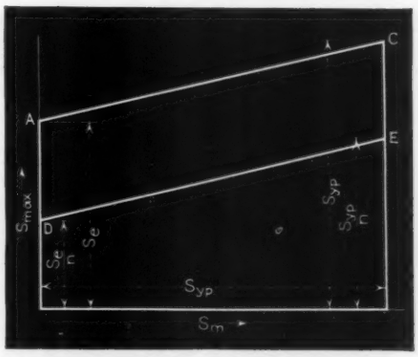


Fig. 2—Graphical method of showing combinations of a mean stress (horizontal ordinate) and a variable stress. Maximum stress at any instant is represented by vertical ordinate

of failure. This is a new field of study and for ductile materials it is reasonable to assume, as is done with static loads, that failure is a function of the distortion or shear energy. It will therefore be assumed that failure because of combined fluctuating stresses occurs when the shear energy corresponding to the maximum and mean components of stress reach the corresponding values of the shear energy at failure in simple tension. Tests by H. C. Gough, see Report of the National Physical Laboratory, *Engineering*, Vol. 3626, July 12, 1935, p. 43, show that this theory is in good agreement with test observations.

To express mathematically this shear energy theory for fluctuating loads first consider the shear energy at failure in simple tension. It can be shown that the shear energy for a stress S is

$$V = \left(\frac{1+v}{3E}\right) S^2$$

where v is Poisson's Ratio, and E is the modulus of elasticity in tension. The values of the shear energies corresponding to the maximum and mean stresses are

$$V_{max} = \left(\frac{1+v}{3E}\right) S^2_{max}, \text{ and}$$

$$V_m = \left(\frac{1+v}{3E}\right) S^2_m$$

Substituting the values of S_{max} and S_m from these equations in Equation (1), there results

$$\sqrt{V_{max}} = (1-m)\sqrt{V_m} + S_e\sqrt{\frac{1+v}{3E}} \qquad (3)$$

That is, for a shear energy V_m corresponding to the mean stress S_m. Equation (3) will define the values of the maximum shear energy corresponding to the maximum stress at failure.

In the case of combined stresses, let the components of stress at a point in a body be S_x and S_y for the normal stresses and S_s for the shearing stress. As the loads change in magnitude with time, these stress components will also vary and the maximum values of these stresses may be denoted by S'_x, S'_y, S'_s, and the mean values by S''_x, S''_y, and S''_s. It will be assumed that the maximum components of stress occur at the same instant of time and the mean components occur at the same instant of time. This is justified since in most applications the loads are synchronous

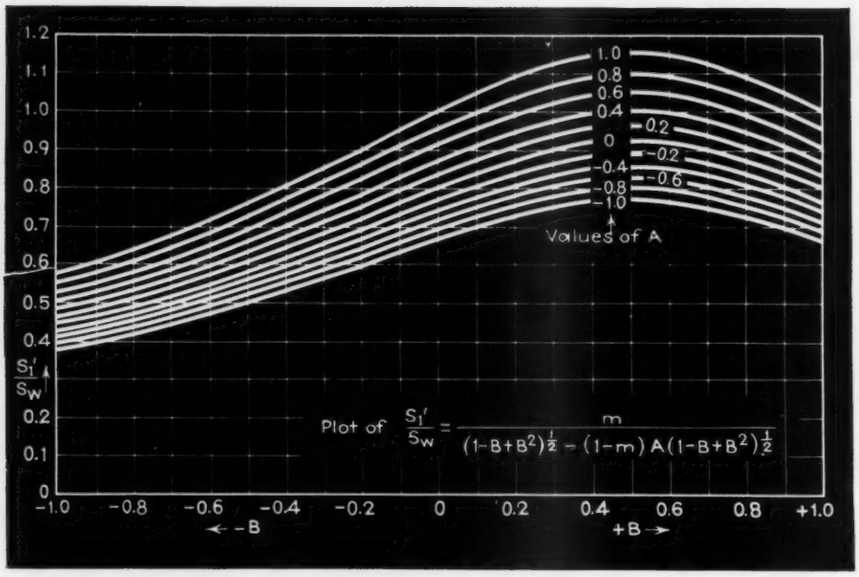


Fig. 3—Graph showing the variation in the maximum combined normal stress as compared to the working stress in simple static tension for different stress ratios

and give such a condition. The shear energy for an element subjected to the stresses S_x, S_y, *and* S_s is

$$V = \left(\frac{1+v}{3E}\right) K_1 \text{ ; in which}$$

$$K_1 = (S^2_x - S_x S_y + S^2_y + 3\,S^2_s)$$

The shear energies corresponding to the maximum and mean stress components are from this equation

$$V_{max} = \left(\frac{1+v}{3E}\right) K_2 \text{ ; in which} \qquad (4a)$$

$$K_2 = (S'_x)^2 - S'_x S'_y + (S'_y)^2 + 3\,(S'_s)^2$$

$$V_m = \left(\frac{1+v}{3E}\right) K_3 \text{ ; in which} \qquad (4b)$$

$$K_3 = (S''_s)^2 - S''_x S''_y + (S''_y)^2 + 3\,(S'')^2$$

It will now be considered that failure occurs when the values of the shear energy V_{max} and V_m reach the values at failure for simple tension as given in Equation (3). Then substituting these values from Equation (4) in Equation (3), there results the expression

$$S_s = \sqrt{K_2} - (1-m)\sqrt{K_3} \qquad (5)$$

That is, the relation between the components of stress for failure is defined by Equation (5). To determine the relation between the allowable stress components it is only necessary to replace S_e by its allowable value S_e/n where n is the factor of safety. Noting that

$$\frac{S_e}{S_{yp}} = m, \text{ then } \frac{S_e}{n} = \left(\frac{S_{yp}}{n}\right) m = S_w m$$

or from Equation (5) the working stress S_w is

$$S_w = \left[\sqrt{K_2} - (1-m)\sqrt{K_3}\right] / m \qquad (6)$$

In this way is obtained the allowable relation between a given working stress S_w and the stress components.

To show the effect of combined stresses and fluctuating loads on the working stress for simple tension, the allowable value of the maximum stress component will be obtained. Considering the special case where the stress components are the principal stresses, then

$S'_s = S''_s = 0$, $S'_x = S'_1$, $S''_x = S''_1$, $S'_y = S'_2$, and $S''_y = S''_2$

from Equation (6) there results

$$\frac{S'_1}{S_w} = \frac{m}{\left[\sqrt{1-B+B^2} - A\,(1-m)\,\sqrt{1-B+B^2}\right]} \qquad (7)$$

where

$B = S''_2/S''_1 = S'_2/S'_1$, $A = S''_1/S'_1 = S''_2/S'_2$, and $S'_1 > S'_2$

Thus from Equation (7) the allowable value of the maximum stress component S'_1 can be determined for different stress ratios A and B. The variation in the maximum stress S'_1 as compared to the case of simple static tension S_w is shown in Fig. 3 for different stress ratios and for a value of m equal to 0.8 which is an average for medium carbon steels. Inspection of Fig. 3 shows that the working stress can be as low as 40 per cent of that in simple tension.

Applications

Transmission shafting under various conditions of loading can be designed using the foregoing theory. The diameters so obtained will be compared with values obtained by using the A.S.M.E. "Code for the Design of Transmission Shafting," and the Westinghouse Code see C. R. Soderberg, "Working Stresses," Journal of Applied Mechanics, *Trans.*, A.S.M.E., Vol. 57, Sept., 1935, p. A-106. To indicate the effect of the fluctuating stress, a comparison will also be made using the shear theory and assuming the loads as static.

CASE 1—*Rotating Circular Shaft Subjected to Static Pure Bending and Static Torque.*

A rotating shaft subjected to a pure bending moment M and a torque M_t is shown in Fig. 4. For an element at point A there is produced a shear stress S_s by the torque and a normal stress S_x by the bending moment such that

$$S_x = (Mr \sin \alpha)/I \text{ and } S_s = M_t r/J$$

As the shaft rotates the value of S_x varies from a maximum of $+Mr/I$ at B to a minimum $-Mr/I$ at C, while the maximum shear stress remains constant. Then the maximum and mean components of stress are

$S'_x = +Mr/I$, $S''_x = 0$, and $S'_s = S''_s = Mr/J$

Substituting these values in Equation (6) and solving

$$D_1 = 2r = k\left[\sqrt{4R^2+3} - (1-m)\sqrt{3}\right]^{\frac{1}{3}}\left[16/(\pi m)\right]^{\frac{1}{3}} \qquad (8)$$

where

$k = (M_t/S_w)^{\frac{1}{3}}$ and $R = M/M_t$

By the Westinghouse Code it can be

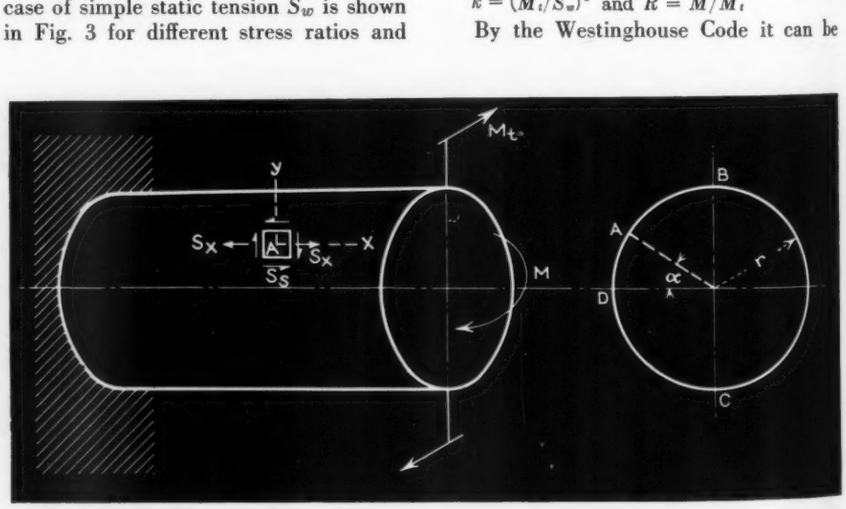


Fig. 4—Stresses on an element in a rotating shaft which is subjected to a bending moment and a torque

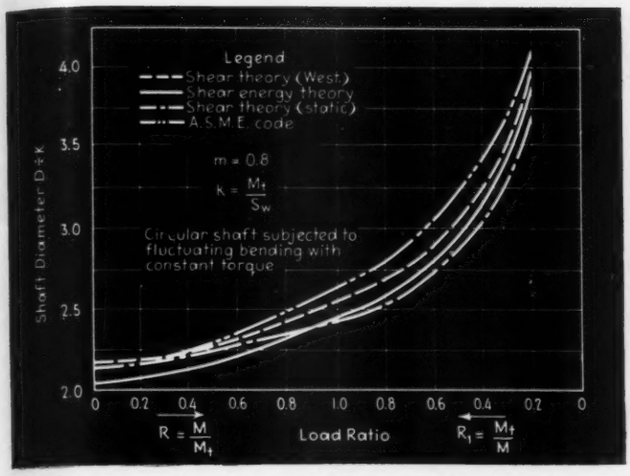


Fig. 5—Diameter of a shaft, under fluctuating bending with constant torque, at various load ratios as calculated by four different codes referred to in legend

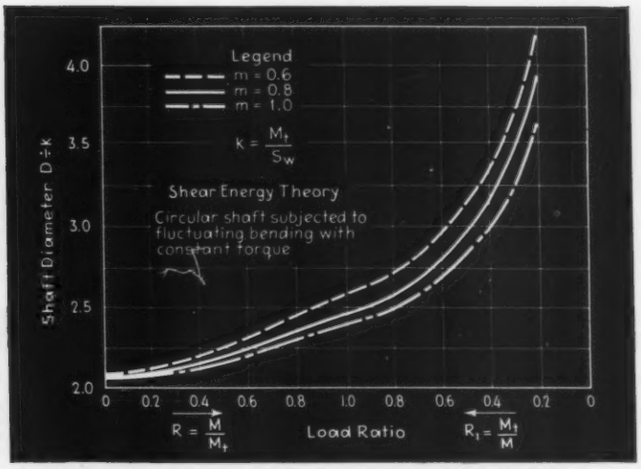


Fig. 6—Required diameter of shaft, under fluctuating bending with constant torque, for different load ratios as calculated by proposed shear energy theory

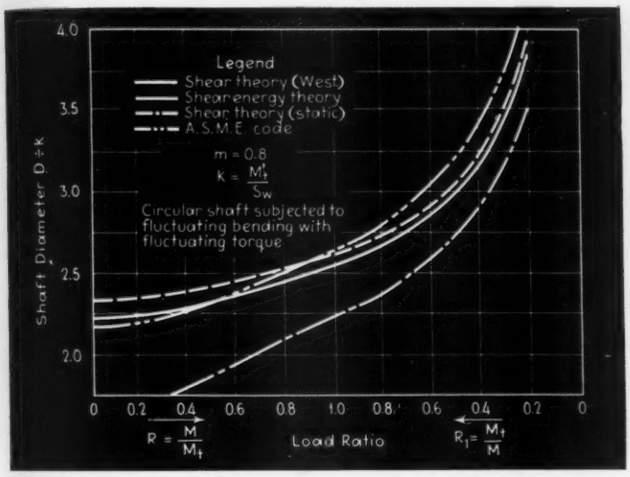


Fig. 7—Diameter of shaft, under fluctuating bending with fluctuating torque, at various load ratios as calculated by four different codes referred to in legend

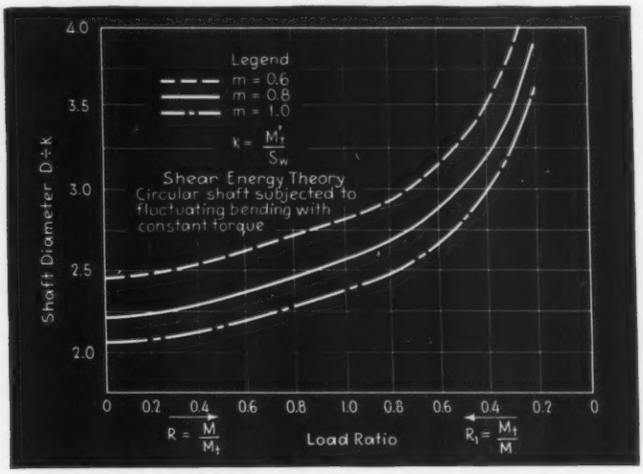


Fig. 8—Diameter of shaft, under fluctuating bending with fluctuating torque, required for different load ratios as calculated by proposed shear energy theory

shown that the required diameter for a given load ratio R is

$$D_3 = k\left[\sqrt{4R^2 + 4m^2}\right]^{\frac{1}{3}} (16/\pi m)^{\frac{1}{3}} \quad (9)$$

Using the A.S.M.E. Code for the design of Transmission Shafting the diameter is by the above notation

$$D_3 = k\left[\sqrt{(1.5R)^2 + 1}\right]^{\frac{1}{3}} (32/\pi)^{\frac{1}{3}} \quad (10)$$

Considering the loads as static and using the maximum shear theory the diameter is

$$D_4 = k\left[\sqrt{4R^2 + 4}\right]^{\frac{1}{3}} (16/\pi)^{\frac{1}{3}} \quad (11)$$

In Fig. 5 the variation in the diameter with change in load ratio R is shown for each of the foregoing methods of design. The curves are plotted for an average value of m equal to 0.8. An inspection of this diagram shows that there can be considerable difference in a design depending upon which code is used. Fig. 6 represents the diameter of shaft required for different load ratios and for different values of m using the proposed shear energy theory.

CASE 2—*Circular Shaft Subjected to Fluctuating Pure Bending and Fluctuating Torque.*

Shafts are sometimes subjected to both a fluctuating bending moment M and a fluctuating torque M_t. Let the values of these loads vary from maximum values of M' and M_t' to minimum values of M'' and M''_t. It will be assumed that the maximum and minimum values of these loads occur at the same instant of time. Then the maximum and mean components of stress are

$$S'_x = +M'r/I, \quad S''_x = 0, \quad S'_s = M'_t r/J,$$

and $$S''_s = \left[(M'_t + M''_t)/2\right](r/J)$$

Using these values of stresses in Equation (6) the diameter required by the shear energy theory is

$$D_5 = k'\left[\sqrt{4R^2 + 4}\right]^{\frac{1}{3}} (16/\pi m)^{\frac{1}{3}} \quad (12)$$

where

$k' = (M'_t/S_w)^{\frac{1}{3}}$ and $R = M'/M'_t$

By the Westinghouse Code the diameter required is

$$D_6 = k'\left[\sqrt{4R^2 + m}\right]^{\frac{1}{3}} (16/\pi)^{\frac{1}{3}} \quad (13)$$

Using the A.S.M.E. Code

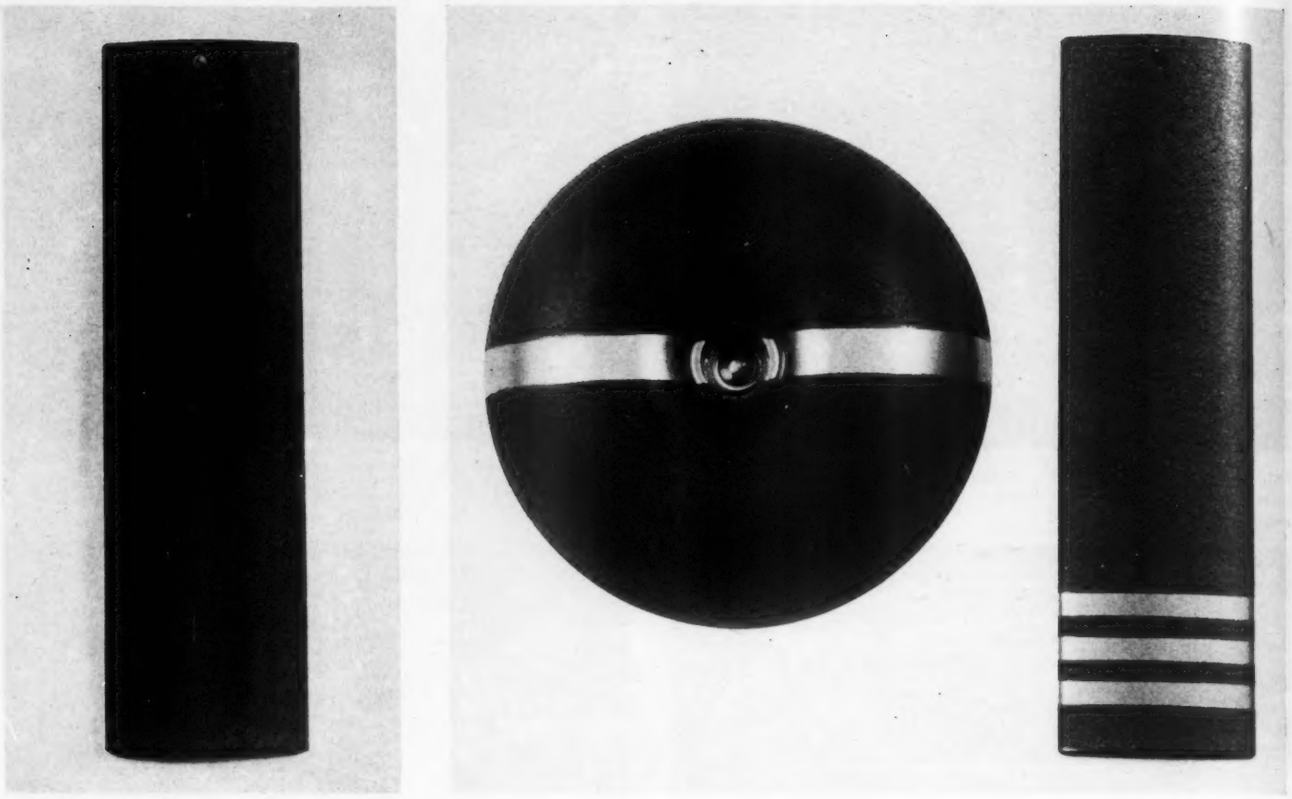
$$D_7 = k'\left[\sqrt{4R^2 + 3}\right]^{\frac{1}{3}} (16/\pi m)^{\frac{1}{3}} \quad (14)$$

Considering the stresses as static and using the shear theory the required diameter is

$$D_8 = k'\left[\sqrt{(1.5R)^2 + 1}\right]^{\frac{1}{3}} (32/\pi)^{\frac{1}{3}} \quad (15)$$

A comparison of the diameter required by the above codes for different load ratios is shown in Fig. 7 for a value of m equals 0.8. Fig. 7 shows that there may be considerable difference in the diameter of the shaft depending upon which code is used. In the case of the A.S.M.E. Code for all load ratios it is on the safe side as compared to the proposed method, the maximum difference being about 7 per cent. Fig. 8 represents the shaft diameters required for various values of m using the proposed shear energy theory.

NOTE: While test results are few, agreement is best with the proposed shear energy theory.



TEXTURED ENAMELS

For Improved Wrinkle Finishes

A. MIESEM
Roxalin Flexible Lacquer Company, Inc.

A new kind of enamel that possesses all the utilitarian advantages but not the drawbacks usually associated with wrinkle finishing materials previously available

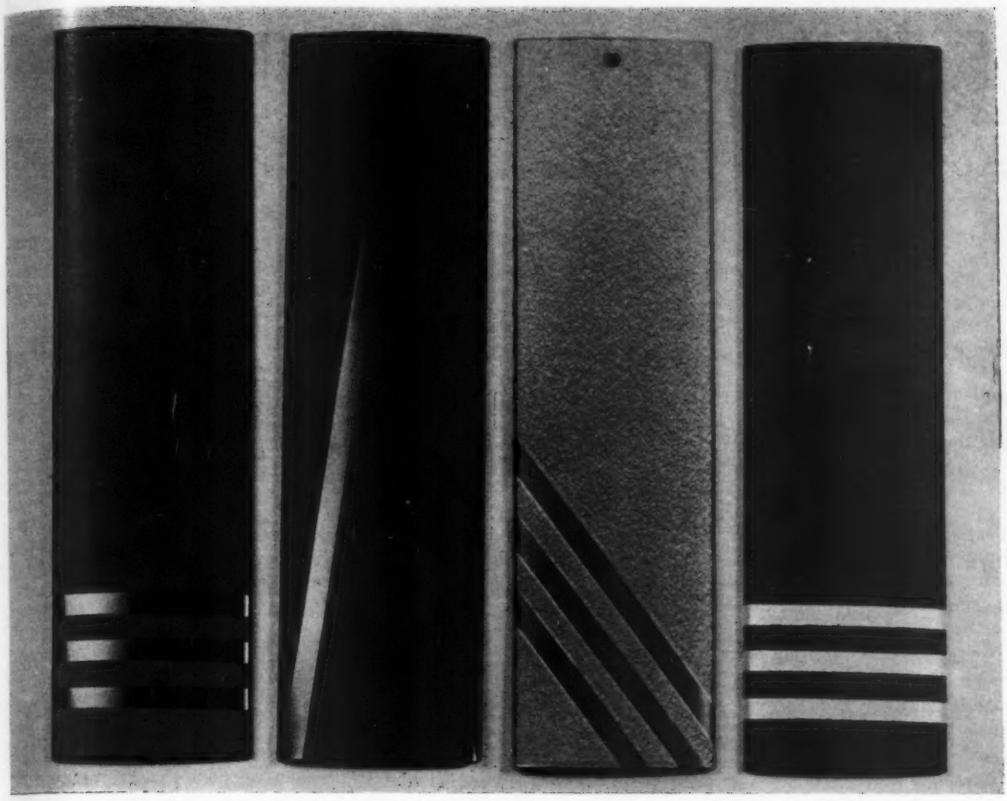
WRINKLE FINISHES originally gained their popularity by their ability to conceal metal defects, spot welds and surface imperfections, in a one-coat application, whereas previously several applications had been necessary with the usual surfacing and top coats. To obtain the economies thus made possible, however, users were faced with a high proportion of rejections for finish faults, sometimes up to 30 per cent of production, owing to the extremely uncertain performance of these wrinkle materials. Such rejected pieces have had to be stripped or washed to the bare metal and completely refinished. This costly experience is common to all users of conventional wrinkle enamels.

Another difficulty with the conventional wrinkle finishes was caused by the extremely sensitive nature of the finishing materials, they frequently bodying or "livering" in the container to a point where they could not be reclaimed. In partly filled containers they rapidly formed a thick skin which could be removed, but not without affecting the properties of the enamels. And also on account of these characteristics they had to be sprayed with constant vigilance as slight variations in film thickness caused differences in appearance. When applied, they often have had a tendency to slip down on vertical surfaces, especially after entering the bake oven, which resulted in fatty edges, sags and "pine trees."

Again, because conventional wrinkle enamels are generally employed for reasons of utility and theoretical economy, their decorative possibilities have not been fully explored. In fact, many finishes of this type, having a coarse, uneven pattern and a tinny gloss, are completely lacking in eye appeal.

These drawbacks to the use of conventional wrinkle finishes, with all their inherent desirable qualities, constituted a challenge that paint chemists could not ignore. As a result, it is now possible to obtain a new kind of textured enamel, which offers all the utilitarian advantages afforded by wrinkles, together with freedom from the production-choking tendencies of these earlier finishes and also possessing high decorative values. These finishing materials posses the following characteristics:

1. Ability to produce a perfectly uniform appearance on all surfaces of the finished product, without glossy spots,



LEFT TO RIGHT

Fig. 1—The lower half of this panel was resprayed to show how the damaged finish can be repaired without destroying uniformity

Fig. 2—Even on surfaces as different as those of the aluminum die-casting and steel panel shown here, perfect matching is readily obtained with the new wrinkle enamels

Fig. 3—Textured enamels can be set off with bright trim of smooth baked enamel in the same color or in a harmonizing tone

fish eyes or fatty edges, independent of the skill of the individual operator.

2. Uniformity of appearance on different parts, when assembled after finishing, even though sprayed by different men.

3. Uniformity of appearance on various surfaces of different character, for example, cast metal and sheet steel. Fig. 2 shows an aluminum die-casting and a steel panel finished in the same material, and having a perfectly matched appearance.

4. Perfect pattern or texture formation, whether the finish is baked immediately or allowed to stand in the air for several hours before entering the heat.

5. Repairability of the film if it should be damaged by coming in contact with another part during the finishing process or through being touched, as by a worker's garment. Instead of requiring complete removal to the bare metal, the new type textured enamels may be sanded in the damaged area and re-sprayed, with the assurance of perfect uniformity. The same appearance results from one coat or two coats. Fig. 1 shows an example of extreme finish damages and subsequent repair.

6. True decorative value, obtained through close-knit textures and soft, glare-absorbing sheens, without any sacrifice of the defect-concealing function of the finish. Such patina luster is not only attractive but is mandatory on products like business machines, where it eliminates eye strain.

These textured enamels, in subdued luster, can be set off with a bright trim of smooth baked enamel in the same color or in a harmonizing tone. Fig. 3 suggests how effective such treatment can be from the appearance standpoint. More important to the production man, it is also quick and economical. The smooth enamel—a special quick-drying type—is applied only in the areas where the trim is to show and after a short drying period is protected by a mask or template while the textured enamel is being applied. The two materials are then baked out together.

7. And finally, serviceability. The close knit structure of these new textured finishes minimizes dirt collection and makes cleaning easier. Based on scientifically designed and controlled vehicles, they are remarkably resistant to marring, moisture, perspiration and abrasion.

Properly baked, the finish is nail hard and cannot be damaged except by mechanical means. The film does not soften even after 500 hours exposure to 100 per cent humidity at 100 deg. F. Constant bodily contact in humid weather proves this type finish the equal of any organic base enamel in perspiration resistance. Perspiration, transferred by handling, is a powerful, if little recognized, factor in the destruction of protective coatings. And, where the conventional wrinkle enamels wear through in 35,000 strokes of a typical abrasiometer, the new textured finish withstands 150,000 strokes, without complete failure.

Furthermore, it is now possible to obtain touch-up wrinkle enamels, in textures matching production finishes for use by distributors and dealers in the field. The touch-up problem has always been a difficult one, but these newly perfected materials may be applied by small portable spray guns, or even carefully brushed in small patches, and hardened in one hour or so by the application of localized heat. A portable therapeutic heat lamp with a 260 watt bulb, supplies ample heat for this purpose.

Uniformity of appearance is also of value where a number of different parts, each sprayed individually, are to be assembled. One company reports that they have as many as 60 different operators spraying parts which are subsequently assembled without any differences in finish being discernible.

Designers in industry seem to be increasingly conscious of the possibilities held by this attractive dull-textured effect, particularly when it is set off with a bright trim in the same color or one which is harmonious. In their present state of development, these finishes often offer a practical solution to the question, "What can we do with this well-designed product to give it that final touch or sales appeal?"

FORCED VIBRATIONS

And Resonant Frequencies in Machinery

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Part I—Outline of the energy method for finding resonance frequencies of machine parts and a procedure for estimating the frequencies of systems having various degrees of freedom

FORMERLY a rather special subject, the study of vibrations is becoming one of everyday importance to designing engineers. Two trends of design are responsible for this growing interest. Big machines and the economical use of strong materials have brought down resonance frequencies into the range of high speed operation which is now demanded. In designs where the presence of resonance frequencies is not expected and provided for, the results sometimes are annoying and possibly destructive. Moreover, an understanding of the mechanism of vibrations will show the advantages which can be obtained by intentional operation near the resonance frequency. While this trick has been employed for hundreds of years in the mechanisms of musical instruments and timepieces, it is now applied to an increasing number of devices such as balancing machines, screens, and testing machines.

The first and often the only question in the study of a vibration problem concerns the resonance frequency of the system. Three methods are available for finding the answer to this question.

One may examine the forces and moments acting in the system and the accelerations which they produce. This procedure leads to a differential equation which sometimes has a general solution, but more often not. Usually laborious step by step integrations must be used to obtain particular solutions. This is the classic method, and while it is too intricate for general design work it is, however, the precision instrument which made possible the development of the two other methods, both of which are based on the assumption that the time-displacement charts of most engineering vibrations are approximately sine curves. Naturally the solutions obtained are correct only as far as this assumption is true. The rare exceptions for which the assumption is not good enough will be mentioned in the final section of this article.

For electrical vibration problems such as occur in alternating current and radio engineering, a symbolic method is widely used. Similar methods may be applied to mechanical vibration problems and will be useful to those who are either familiar with alternating current calculations or to those who deal with one type of vibrations so frequently that it is worth their time to develop a special calculating technique. Such special methods are used for crankshaft vibration analysis.

The simplest and most far-reaching tool with which most mechanical vibration problems can be solved with relative ease is the energy method which was first developed by Lord Rayleigh in his Theory of Sound (1877). This method has the special advantages of being elementary, of including conditions which might otherwise be overlooked, of giving good approximations even when the shape of the vibrating system is not known and of involving only a few mechanical notions.

This article will deal with the energy method of finding resonance frequencies. An outline, examples of various types and some notes are given with the aim of enabling engineers who are not familiar with frequency problems to estimate rapidly the frequencies of proposed designs and also to change effectively the frequencies of existing systems.

The notions which are used in the energy method are: the sine-curve assumption (often called harmonic motion), the potential energy and the kinetic energy in the system. These and the component notions of center of gravity, moment of inertia, spring rate and mass can be found in any engineering hand book and need not be reviewed here.

The Energy Method

As an example of a vibration, consider a common rope swing. In this system, energy continuously changes in form from potential to kinetic and conversely. The potential energy shows in the height above the lowest point and the kinetic energy in the speed. Observation shows that at the highest point, where the potential energy is maximum, the speed and the kinetic energy are both zero. Speed then increases and height decreases until the lowest point is reached where speed is maximum and the potential energy is zero. From there on the exchange continues in inverse direction, speed decreasing and height increasing.

If the air resistance is neglected, it can be said that no energy is added to or taken from the swing on its course and that the sum of kinetic and potential energy must be constant. As far as the frequency is concerned, the air resistance has no effect and may be neglected.

For a detailed analysis of the system, the varying strain on the rope, the air resistance and the acceleration at each moment should all be considered. If the frequency of the system and the manner of applying the energy method to find it only are of interest, then it is necessary only to consider the maximum potential energy, that is, the work necessary to lift the swing to the highest point of its path, and the maximum kinetic energy, that is, the work stored at maximum speed or the product (½ mass multiplied by the square of speed) for all the moving parts. The potential energy at the highest point (or at full amplitude) is entirely independent of the frequency. The kinetic energy is proportional to the square of the maximum speed which in turn is proportional to the frequency. These two energies are equal and by equating them the frequency can be found.

In a complicated system, the several parts will have different amplitudes and the relation of these amplitudes to each other is often hard to find. In such systems any reasonable assumption about the amplitudes will give a good approximation of the frequency, as will be seen from examples. It can be shown that any wrong assumption will give too high a value for the fundamental frequency. Therefore the approximation found by a wrong assumption can be improved by varying the ratio of the different amplitudes and looking for a minimum of the frequency. Examples will show the process better than it can be described.

For most engineering purposes, this refinement will not be necessary. The data, such as weights of parts, spring rates and stiffness of supports usually contain tolerances which affect the frequency more than the error caused by a wrong assumption. Third decimal precision is often wasted effort.

Before going on to the examples, the constant relation between frequency and maximum velocity shall be established, so that it need not be repeated in all the solutions.

If the frequency is f cycles per min. and the amplitude is a, the average speed per minute is 4 af, because one complete swing goes from $+a$ to $-a$ and back. The average speed per second is

$$\left(\frac{4}{60}\right)(af) = \frac{af}{15}$$

Assuming that the time displacement chart is a sine curve (which it practically is in most cases) the maximum speed is 0.5 π times the average speed. Therefore the maximum speed

$$V_{max} = \left(\frac{\pi}{30}\right)(af)$$

The relation

$$V_{max} = \left(\frac{\pi}{2}\right)(V_{average})$$

can be shown to exist by considering the sine curve as the projection of a uniform circular motion, by calculus or perhaps most convincingly by plotting out the sine curve, plotting the speed from it, and comparing the maximum speed to the average speed. The process is shown in Fig. 2 because the relation is fundamental.

EXAMPLE *1. Systems with one degree of freedom.*

A setup like the one shown in Fig. 3 was used to find the radius of gyration of an automobile from the measured frequency. Assuming the mass distribution to be known, the frequency is found as follows:

(A) The system can perform only one type of motion: i.e., rocking about the pivot. It has only one degree of freedom. With the assumption of a maximum angular deflection of α radians, the configuration of the system is completely defined.

(B) Assuming α downwards, the work E necessary to push the car down to the position of maximum deflection can be found as follows:

$\alpha\, l$ = max. compression of spring
$\alpha\, l\, c$ = max. force of spring
$\alpha\, l^2 c$ = max. moment of spring about pivot
$\alpha\, h$ = outward shift of center of gravity
$W\, \alpha\, h$ = increase in moment helping to compress the spring
$\alpha\,(l^2 c - Wh)$ = max. moment resisting spring compression

Since the moment increases in proportion to α

$(1/2)\,[\alpha\,(l^2 c - Wh)]$ = average moment resisting spring compression, and
$(1/2)\,[\alpha^2\,(l^2 c - Wh)]$ = deflection multiplied by average resistance = the total work E

(C) The maximum kinetic energy E, expressed with the unknown frequency f, is found as follows:

$\left(\frac{\pi}{30}\right)(af)$ = max. angular velocity per sec.

i = radius of gyration

$\left(\frac{W}{g}\right)(i^2)$ = mass moment of inertia around c. g. axis

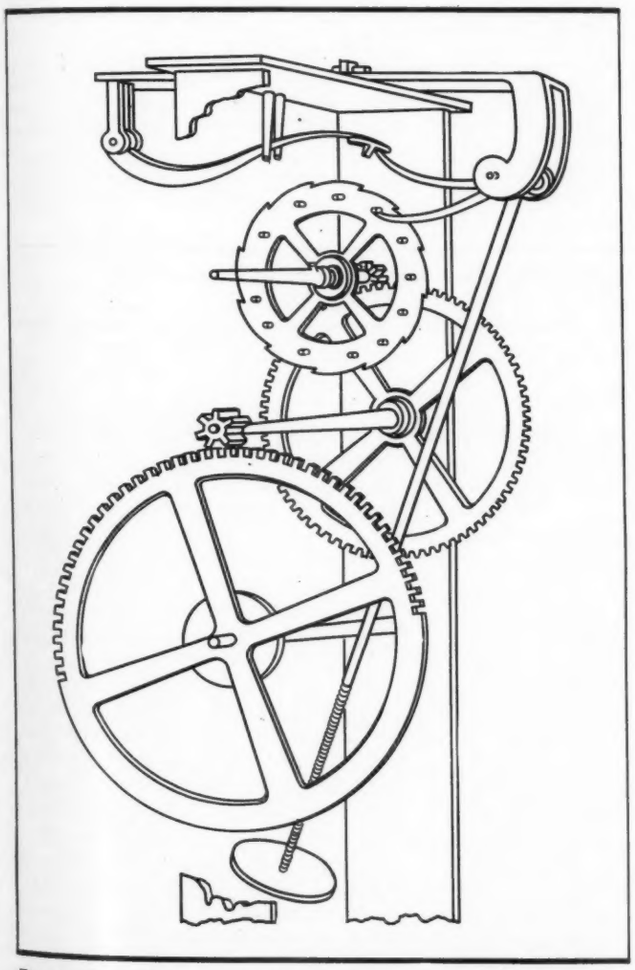


Fig. 1—An early example of a resonance machine. This illustration is taken from a drawing made about the end of the sixteenth century by Galileo for a pendulum clock

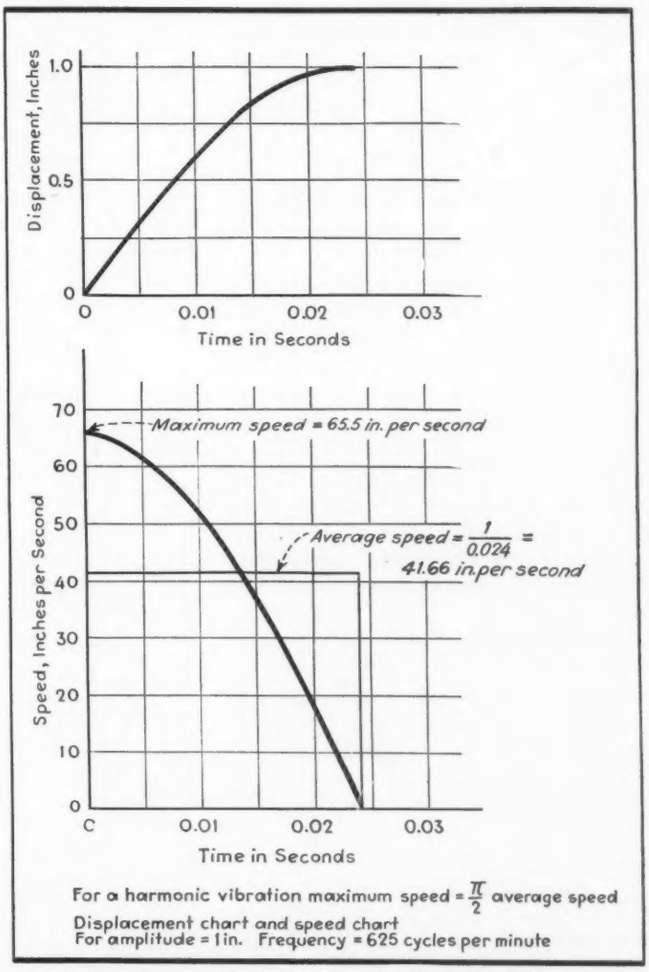


Fig. 2—Graphical method of comparing the relation between the maximum speed and the average speed of a motion the time displacement chart of which is a sine curve

$\sqrt{e^2+h^2}$ = distance of c. g. from pivot

$\left(\frac{W}{g}\right)(i^2+e^2+h^2)$ = mass moment of inertia around pivot

$\left(\frac{\pi}{30}\right)^2 (af)^2\left(\frac{W}{2g}\right)\left(i^2+e^2+h^2\right)$ = max. kinetic energy or

$$E = \left(\frac{\pi}{30}\right)^2\left(\frac{W}{g}\right)\left(\frac{a^2}{2}\right)\left(i^2+e^2+h^2\right)f^2$$

(D) Equating maximum potential energy and maximum kinetic energy and solving for the unknown f there results

$$f^2 = \left(\frac{30}{\pi}\right)^2\left(\frac{l^2c - Wh}{i^2+e^2+h^2}\right)\left(\frac{g}{W}\right)$$

and with the values indicated in Fig. 3 the frequency

$$f = \left(\frac{30}{\pi}\right)\left(\sqrt{\frac{12,100\times70-3,000\times34}{2,025+2,500+1,156}\times\frac{386}{3,000}}\right)$$

= 36.6 per min.

This formula is the general form which for special cases reduces to the well-known vibration formulas.

For a pendulum or a scale as shown in Fig. 4 (a) with $e = 0$, $c = 0$, and h negative

$$f = \left(\frac{30}{\pi}\right)\left(\sqrt{\frac{gh}{i^2+h^2}}\right)$$

For the mathematical pendulum as shown in Fig. 4 (b) with $i = 0$ this equation reduces further to

$$f = \left(\frac{30}{\pi}\right)\left(\sqrt{\frac{g}{h}}\right)$$

which if h is given in inches becomes

$$\frac{188}{\sqrt{h}}$$

For a flywheel vibrating in torsion as represented in Fig. 4 (c), with $h = 0$, $e = 0$ and instead of c_1 the torsional stiffness C_1 in inch pounds per radian is used, the general form reduces to

$$f = \left(\frac{30}{\pi}\right)\left(\sqrt{\frac{C_1 g}{W i^2}}\right)$$

Finally for a weight bobbing up and down on a spring as shown in Fig. 4 (d) with $h = 0$, $e = l$ and i negligibly small compared to l, there results by inserting these values in the general form

$$f = \left(\frac{30}{\pi}\right)\left(\sqrt{\frac{cg}{W}}\right)$$

and when c is expressed in lbs. per inch

$$f = \frac{188}{\sqrt{\frac{W}{c}}}$$

Note that in all these formulas the dimensional units must be considered. If h is measured in feet, g must be 32.2 ft./sec.^{2} and all other lengths must also be measured in feet. If the lengths are in inches, g is 386 in./sec.^{2} and all other lengths must also be measured in inches.

Also note that a cancelled out, meaning that the frequency does not depend on the amplitude when the average resistance is one half of the maximum resistance.

The set of formulas may be useful for reference. As most engineering systems are not so simple, but can be attacked in the same manner, the way in which the formulas were obtained is very important. This method of attack will be used in further examples without full explanation and is therefore summarized here:

(A) The kind of motion that the system can perform was determined and an assumption was made concerning the amplitudes.

(B) The work necessary to get the system into the assumed maximum deflection was found, after "restraints," spring forces and location of the weight were considered.

(C) The maximum kinetic energy was found from the assumed amplitudes, mass distribution and unknown frequency of the system.

(D) The energies were equated and solved for the unknown frequency.

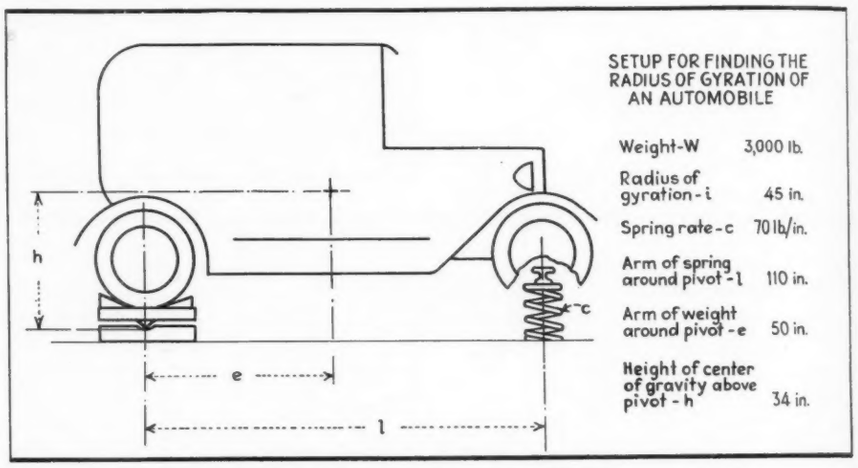

SETUP FOR FINDING THE RADIUS OF GYRATION OF AN AUTOMOBILE
Weight-W 3,000 lb.
Radius of gyration - i 45 in.
Spring rate - c 70 lb/in.
Arm of spring around pivot - l 110 in.
Arm of weight around pivot - e 50 in.
Height of center of gravity above pivot - h 34 in.
h
c
e
l

Fig. 3—Setup for finding the radius of gyration of an automobile

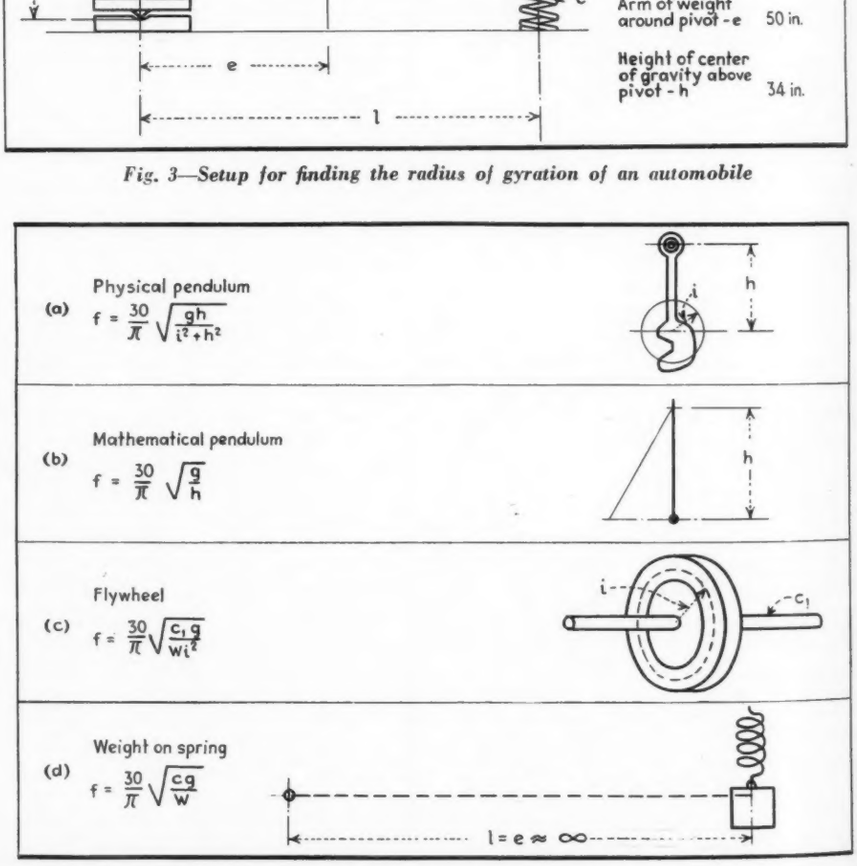


Fig. 4—Some vibration formulas developed for handy reference from the general form for particular cases: (a) Physical pendulum, (b) Mathematical pendulum, (c) Flywheel, and (d) Weight on a spring

WHAT IS NOT INVENTION—III

H. A. TOULMIN, JR.
Toulmin & Toulmin

This third article in the series discusses improvements upon existing patents: omission, duplication, or substitution of parts, increased production speed, adjustability, and their patentability as new invention

TOO MUCH emphasis can hardly be placed on the ultimate test of invention, which is: an invention to be patentable must involve imaginative, inventive genius—not merely the exercise of ordinary mechanical or technical skill. Failure to understand this distinction makes many products of the design room unpatentable, or worse still, gets a company involved in expensive patent suits. The following types of mistake, along this line, are common.

Omission of Parts

Simplifying a machine by omitting parts is not patentable unless it brings out a new result, or one unexpectedly varying in quantity, even though it makes the product cheaper or more convenient. In other words, patentability is not related to, or dependent upon, the number of parts in a device nor the number of steps in a process.

For instance: A patented beer supply system consisted of an upper ice-box connected by a supply pipe with the barrel of beer underneath. To keep the beer in the supply pipe cool the patent specified that the pipe was to be enclosed in a metallic case and the space between the pipe and case filled with sand or gravel through which the drippings from the upper ice-box would trickle.

A competitor tried to avoid this patent by eliminating the sand or gravel so that the cold air from above as well as the drippings would pass down around the pipe and so increase the cooling effect. It was well-known that cold air tends to sink. Therefore the court held that omission of the filler accomplished no new or useful purpose, but merely applied common knowledge and so was not patentable invention.

In connection with a case involving a hat pouncing machine the court said: "The case . . . resolves itself into the question whether the omission of the feed roll involves invention; and in view of the fact that the hat support and pouncing cylinder of the Eickemeyer patent will accomplish practically the same function as the Taylor, though not so perfectly, we hold it does not—in other words, it required no invention to omit the feed roll of the Eickemeyer patent, and to make the subsidiary changes necessary to produce a working device."

It seems worth-while here to give one example in which omission of a part constituted patentable invention. The court said: "The machinery and apparatus used by Lawton had all been used before. His only discovery was an improvement in the process. He found that, by altogether omitting one of the steps of the former process—the grinding and mixing under the muller stone —and mixing in the mixing machine by means of steam, a great improvement was effected in the result.

"Why should it be doubted that such a discovery is patentable? It is highly useful, and it is shown by the evidence to have been the result of careful and long-continued experiments and the application of much ingenuity."

Duplication of Parts

Neither mere multiplication of parts, nor multiplication of results therefrom, constitutes patentable invention. One case will suffice. The court said in part: "(The invention) consists merely in putting an additional pane of glass in the fare-box opposite the side next the driver, so that the passengers can through it see the interior of the box. Such a contrivance does not embody or require invention. It requires no more invention than the placing of an additional pane of glass in a show-case for the display of goods, or the putting of an additional window in a room opposite one already there. It would occur to any mechanic engaged in constructing fare-boxes that it might be advantageous to insert two glass panes —one next the driver and the other next the interior of the car. But this would not be invention within the meaning of the patent law."

Whether a device is extremely simple or highly complex has nothing whatever to do with its patentability. Yet many overlook this fact and attempt to get a patent on a machine merely because it is complex and consists of a multiplicity of parts.

This subject is so closely allied to that of combinations and aggregations that it will be discussed under that heading.

Increased Production Speed

In the past few decades some of the most industrially important developments have been in securing greater speed or production. The law covering this seems to be quite confusing to many laymen. Generally speaking, such a development is held to be the expected result of the ordinary activities of engineers, designers and mechanics to increase production speed, and is therefore not patentable. The Supreme Court has said: "It is hardly necessary to say that it is no invention, within the meaning of the law, to perform with increased speed a series of . . . operations old in themselves, and in the order in which they were before performed."

This rule carries over, to some extent, into the field of machines which perform, either automatically or semi-automatically, operations which were formerly done by hand—which of course always increases the speed of production.

But it has also been held that if a machine produces a product of a quality which could not be attained by handwork, it is patentable.

Also, the patentability of a machine may depend upon the degree of the speed increase. It is up to the court to decide whether the increase in speed is merely that which would be expected from the ordinary advance in engineering knowledge or the natural slow evolution of an art, or whether it is so great as to be far beyond anything that might normally be expected.

The line is oftentimes a fine one, so I shall quote a few court decisions to show the reasoning that can be expected of the courts when the patentability of a machine depends on its increased production speed.

The Supreme Court has said: "It may be laid down as a general rule, though perhaps not an invariable one, that if a new combination and arrangement of new elements produce a new and beneficial result, never attained before, it is evidence of invention. It was cer-

tainly a new and useful result to make a loom produce fifty yards a day when it never before had produced more than forty; and we think that the combination of elements by which this was effected, even if those elements were separately known before, was invention sufficient to form the basis of a patent."

In the hat pouncing machine case, referred to in another connection, the court said apropos of speed: "It is true that the Taylor machine seems to be capable of doing more work and at less expense for labor and pouncing material than the prior devices, which it appears to have largely supplanted; but this consideration, while persuasive, *is by no means decisive, and is only available to turn the scale in cases of great doubt respecting the validity of the invention.*" I have italicized the last words of the quotation because that seems to be the essence of the rule governing increase in speed: namely, that it is not invention of itself, but may be considered in resolving doubtful cases. The same general idea has been expressed several times by the Supreme Court, as in the Eibel paper machine case where increased speed proved invention.

Substitution of Equivalents

The common store of mechanical and technical knowledge contains a vast number of principles and devices which, in their effects, are equivalent. This is especially true of mechanical movements, electrical parts and circuits, and prime movers. Thus a weight and a spring are equivalent. In many cases it makes no difference whether a piece of equipment is driven by a steam engine, a gasoline motor or an electric motor. While it is true that the substitution of one device or principle for another may improve the result, if the performance is substantially the same there is no patentable invention.

In one case the court held: "Authorities concur that the substantial equivalent of a thing, in the sense of the patent law, is the same as the thing itself; so that if two devices do the same work in substantially the same way, and accomplish substantially the same result, they are the same, even though they differ in name, form or shape. Apply that principle to the case before the court, and it is clear that the knife in the respondents' machine, when considered in connection with the striker, is substantially the same thing as the cutter in the machine of the complainants when put in operation by the means employed to raise it and let it fall to perform the cutting function, without which the machine would be of no value."

In another case the court said: "But in this . . . patent . . . there is nothing novel, either in the pawl or the ratchet, or the mode in which they jointly cooperate to produce the desired result. The fact that the pawl is described as acting substantially at right angles to the ratchet or notched hub does not seem to introduce any new or useful element. The combination covered by the claim . . . is in law and in fact merely a mechanical equivalent for that which was already covered by the Campbell patent, which had the priority of invention."

Adjustability

To apply to an existing machine devices which make it adjustable, or more accurate, or more convenient, or easier to operate or more adaptable to varying operating conditions is within the expected scope of the routine duties of a designer, engineer or mechanic, and does not call for inventive genius.

The court said in one case: ". . . supplying the feature of the truss rods and the adjustable stay rods did not constitute invention. To provide such supports would be but a step obvious to any skilled mechanic."

Again it was held: ". . . (the claims) introduce into the otherwise well-known combination of elements open to public use merely the feature of a carriage for the negative frame, which carriage is adjustable to accomodate different sizes of such frames. The adjustability of a part does not involve invention."

The next article in this series will discuss, from the point of view of the designer and engineer, the non-patentability of increased durability, convenience and portability, the selection and use of materials.

Making a machine run with a fewer number of parts does not constitute invention

SHAFTLESS MOTORS

Mounting Tolerances, Ratings and Ventilation

R. J. OWEN
The Louis Allis Company

Precautions to be observed and provisions to be made in order to gain the full advantage of safe economy and the maximum drive efficiency possible when the machine spindle also serves as the motor shaft

WHEN THE ROTOR of a motor is to be mounted directly on the work spindle or the driving shaft of the machine the design problem is fundamentally the same whether it be a frame type or a rolled shell type of shaftless motor. The electrical characteristics of the two are essentially the same as those of the general purpose squirrel cage motor. Structurally the frame type is similar to the general purpose motor with bearing brackets, bearings and shaft omitted. It may have feet for base plate mounting, or as is more usual, without feet for end mountings. A machined rabbet is provided on the ends of the stator frame for accurately aligning the motor with the driven spindle. The alignment of the rolled shell type motor depends entirely upon the fit of the stator shell in the housing provided in the machine.

Whether a frame type or a rolled shell type motor will be used depends upon the relative importance of appearance, compactness, weight and other factors, both types possessing the important advantage inherent in shaftless motors, namely, the motor power developed goes directly to the work spindle. In certain high speed and portable motor driven machines such as portable grinders, polishers, sanders, nut setters and hoists the rolled shell type is usually selected because size and weight reduction is of utmost importance. Then, too, where appearance and compactness is wanted, as in a radial drill head, a rolled shell type motor, which can be built to conform to the lines of the machine as well as eliminating auxiliary mechanical transmission equipment, may be used. In many larger machines such as wood planers, joiners, lathes, grinders and similar equipment the frame type motor is usually preferred because the essential requirement is a short rigid driving connection. Appearance and weight are secondary factors and the added difficulties and expense of incorporating a motor into the machine housing are seldom warranted.

Whichever type of shaftless motor is used, variations in the amount of air-gap resulting from eccentricity of spindle axis and housing bore will cause a varying electrical unbalance and in extremes will cause the rotor to strike the stator. For that reason it is important that the tolerances for the bore of the rolled shell motor stator housing, or the shoulder fit for a frame type motor, be specified within close limits. Limits for rolled shell motors should be such that the variation from uniform air-gap shall not exceed ten per cent of the normal air gap as given in Table II. Easy removal of the stator should be insured by allowing 0.001 to 0.002 in. clearance in the bore in the machine frame or housing with proper allowance for the fact that stator shells are ground to a tolerance of plus 0.000, minus 0.002 in. in all sizes.

Rotors of the shell type motors are available with either straight or tapered (¼ in. per foot) bores for various conditions of mounting. The shaft on which the motor is mounted should be ground 0.0005 to 0.001 in. oversize for a satisfactory press fit. Table II gives normal rotor bores for standard motors.

To maintain a uniform air-gap around the circumference of the rotor, the shaft of the driven machine must be designed with sufficient rigidity to carry the weight of the rotor without excessive deflection. The permissible variation of ten per cent in air-gap, previously referred to, must not be exceeded when selecting a shaft according to its deflection characteristics based on the complete shaft with all of the loads and bearings as a unit. Usually the rotor of the frame type motor is overhung in order to obtain maximum space economy and is also located as near as possible to the inner bearing. Where the design prevents the rotor from being located sufficiently close to the machine bearing an outboard bearing is incorporated to keep deflection within permissible limits. This becomes particularly desirable were higher horsepower motors with large rotors and low speed ratings with small air-gaps are used, the outer bearing installation usually being more economical than using a shaft large enough to avoid high deflection. The long length of the rolled shell motor usually requires a bearing at both ends of the rotor.

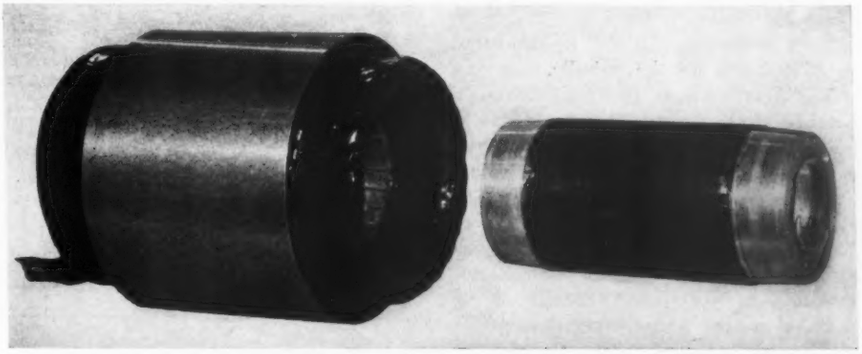


Fig. 1. Rotor and stator of typical rolled shell type shaftless motor. Rotor is of conventional squirrel cage construction. Stator laminations are assembled within the accurately finished steel shell. Construction shown here is typical for diameters of 8,000 in. or less

Proper balance of the assembled shaft and rotor is essential so that deflections caused by dynamic loads do not result in variations in air gap that exceed the permissible variation of 10 per cent of the normal air gap. It is therefore necessary that the assembled rotor and machine shaft with all other parts that are mounted on it such as collars, pulleys, and fans be given a final dynamic

balance. If this final balancing requires the removal of weight at some point on the rotor the material should be removed from the iron core by drilling rather than by modifying the end ring, which latter might affect the performance. Where the mass of the shaft with the part it carries is relatively large it is preferable to remove or add weight on the ventilating fan or balancing collar.

Balancing will also affect the critical speed of the motor. While vibration may be reduced in high speed drives because of absence of pulleys, belts, couplings, overhung brackets and shafts, it may happen that the inertia of the assembled rotor and spindle is such that the operating speed of the motor will be near the critical speed of the shaft. This condition seldom occurs in ordinary practice with rolled shell type motors in the sizes being offered at the present time, as the critical speed will tend to be above the operating speed. In exceptional cases where the weight and speed of the rotating member is comparatively high the critical speed may occur at or near operating speed. Dynamic balancing should not be attempted at critical speed for the reason that the results obtained are often erratic and unreliable and it is, therefore, advisable to consider one or more of the following steps to correct this condition:

1. Increasing the cross-sectional area of the shaft or spindle between bearings.
2. Decreasing the distance between bearings.
3. A redistribution of weight of the rotating parts on the spindle.

For best results, the final dynamic balance of the assembled spindle should be made as closely as possible to the actual operating speed for best results.

Frame type shaftless motors are available in horsepower ratings from ½ to 200 hp. and for all frequencies, voltages and speeds. Rolled shell motors are available in horsepower ratings of ¼ to 30 hp. in speeds and frame dimensions which are standardized at the figures shown in Table I. Speeds below 900 r.p.m. can be obtained in the larger diameter frames. In addition to the usual electrical characteristics available with any motor such as normal torque—normal starting current, normal torque—low starting current, high torque—low starting current, special purpose ratings such as high torque—high slip, intermittent ratings, reversing and multi-speed ratings are also available.

The rolled shell type of motor lends itself to unusual dimensions such as may be required in self contained drives for feed mechanisms, cross-rail, rapid traverse, clutches and chucks. In these applications controls of the auxiliary motors are interlocked with that of the main drive motor. By use of frequency converters the auxiliary drives at various points on the machine may be driven at different speeds. Where space in the axial direction has been limited, motors with stator diameters up to 18 or 20 in. with correspondingly short lengths have been built. Likewise motors unusually long and correspondingly small in diameter are also used where special design requirements warrant such special construction. However, motors with extremely long cores in relation to the diameter will have somewhat lower power factors and efficiencies. Generally this is of comparatively little importance for the average application.

In standardizing on these motors, recognition has been given to possible varied requirements of the machine builder by providing over-lapping ratings. That is, two or more different horsepower ratings are available on a single diameter. In determining the size of motor required for rapid acceleration services, as in rapid traversing, where WR^2 is an important factor, the absence of gears or pulleys interposed between the motor and driven spindle will permit a smaller motor to be used than would be the case with a conventional gear or belt drive. This is in addition to the gain resulting from the higher efficiency of the direct drive.

An important requirement in the application of any built-in motor is to design it so that a sufficient amount of cooling air will be furnished to the motor. Frame sizes are normally established on the basis of continued duty ratings with a temperature rise of 40 deg. C. with adequately ventilated mountings.

For motors with class "A" installation the temperature rise on the basis of intermittent service is 50 deg. C. for open mountings and 55 deg. C. for totally enclosed mountings. Adequate facilities must be provided in all cases for the entrance of cooling air in amounts required for the various types of service. The machine mounting bracket to which the stator frame of the frame type motor is attached should conform in contour, area and location of air openings to those in the bearing bracket which is normally supplied as a part of the complete motor. In designs for special application where the open type motor is not suitable and a drip-proof, splash-proof or explosion proof motor is desired location and size of ventilation openings in the machine frame must conform to that of the motor.

Essentially the same methods of ventilation as are used for the various types of general purpose motors are applicable to built-in motors. The comparatively large mass and external surface of the machine housing is a factor in cooling the built-in motor, as a considerable amount of heat is transmitted by conduction from the motor to the housing casting. The amount of additional venti-

TABLE I. A.C. Rolled Shell Shaftless Motors
Diameter, Length and Horsepower Rating (Non-Standard)

Stator Shell Dia. (in.)	**Overall Length (in.)*	***Horsepower (Continuous)*			
		3600 rpm.	*1800 rpm.*	*1200 rpm.*	*900 rpm.*
4.703	8	1.5	0.5	0.3	...
5.703	11	2.5	1.5	1.0	0.5
6.468	12	7.5	3.0	2.0	1.5
8.000	18	15.0	7.5	5.0	2.0
10.000	17½	20.0	10.0	10.0	7.5
12.375	19	30.0	25.0	20.0	15.0

* This dimension is approximate and includes the stator coils at both ends.
** Horsepower ratings are for motors wound for 60 cycles, 3- or 2-phase in open mountings with ideal ventilation. These are regarded as the highest horsepower ratings obtainable in keeping with good design for the respective diameters.

TABLE II. A.C. Rolled Shell Shaftless Motors
Normal Rotor Bores and Air Gaps

Stator Dia. (in.)	*Air Gap (in.)*		*Rotor Bore*			
			2 Pole		*4–6–8 Pole*	
	2 Pole	*4–6–8 Pole*	*Std.*	*Max.*	*Std.*	*Max.*
4.703	0.020	0.015	1.0	1.25	1.0	1.25
5.703	0.020	0.015	1.25	1.5	1.25	1.5
6.468	0.025	0.015	1.5	1.75	1.5	1.75
8.000	0.0215–0.035	0.015	1.5	2.0	1.5	2.0
10.000	0.035	0.015–0.018	1.875	2.0	1.875	2.375
12.375	0.050	0.020	2.25	2.75	2.25	2.75

lation needed will depend upon the service for which the motor is used.

The amount of heat dissipated depends upon the volume of air and the difference in temperature between the air and the heated surfaces with which it comes in contact. Experience has shown that a rise of 10 to 15 deg. C. in the temperature of the air during its passage over the heated parts gives best results. For a motor with a 50 deg. C. temperature rise this represents a difference of 35 to 40 deg. C. between the air and the core which is very effective under normal conditions. The amount of air required to maintain this relation is 10 to 15 cu. ft. of air per min. per hp. The air velocity should be maintained at 1,000 ft. per min. or higher.

Air circulation can be obtained by mounting suitable fans on the shaft and properly directing the air over hot surfaces at as high a velocity as possible. The fans employed for cooling built-in motors are usually supplied by the machine builder as their design must generally conform to the motor housing which is a part of the machine. The diameter of the fan, especially for motor speeds of 1,800 r.p.m. and below, should be as large as possible to maintain the proper volume and velocity of air and in practice will work out to be approximately the diameter of the stator shell. At speeds of 3,600 r.p.m. fans should have 8 to 10 blades. With the assistance of the motor manufacturer a choice of fan designs can be made depending upon the space available, the location of the motor with relation to machine parts and the degree of its closure depending upon whether it is a semi-enclosed or protected motor, totally enclosed motor, totally enclosed fan-cooled motor or an enclosed motor separately ventilated by an auxiliary blower.

Stator housings must be provided with ducts as shown in Fig. 2, that will direct the proper volume of air over the core and windings at the proper velocity. The air passage between the housing and motor core should have a uniform area without sharp turns. Inlet to the fan and outlet from the motor should be as large as possible so that no head is lost at those points. A constricted path over the hot surfaces is necessary to maintain the proper velocity.

The foregoing applies generally to the mounting and ventilation of motors which operate continuously at their rated output. Where the motor operates only for short intervals, a plain housing without fans or air ducts can be used. A housing without internal ribs or fans but having a full contact with the stator shell provides the most effective means of transferring the heat to the mass of the housing.

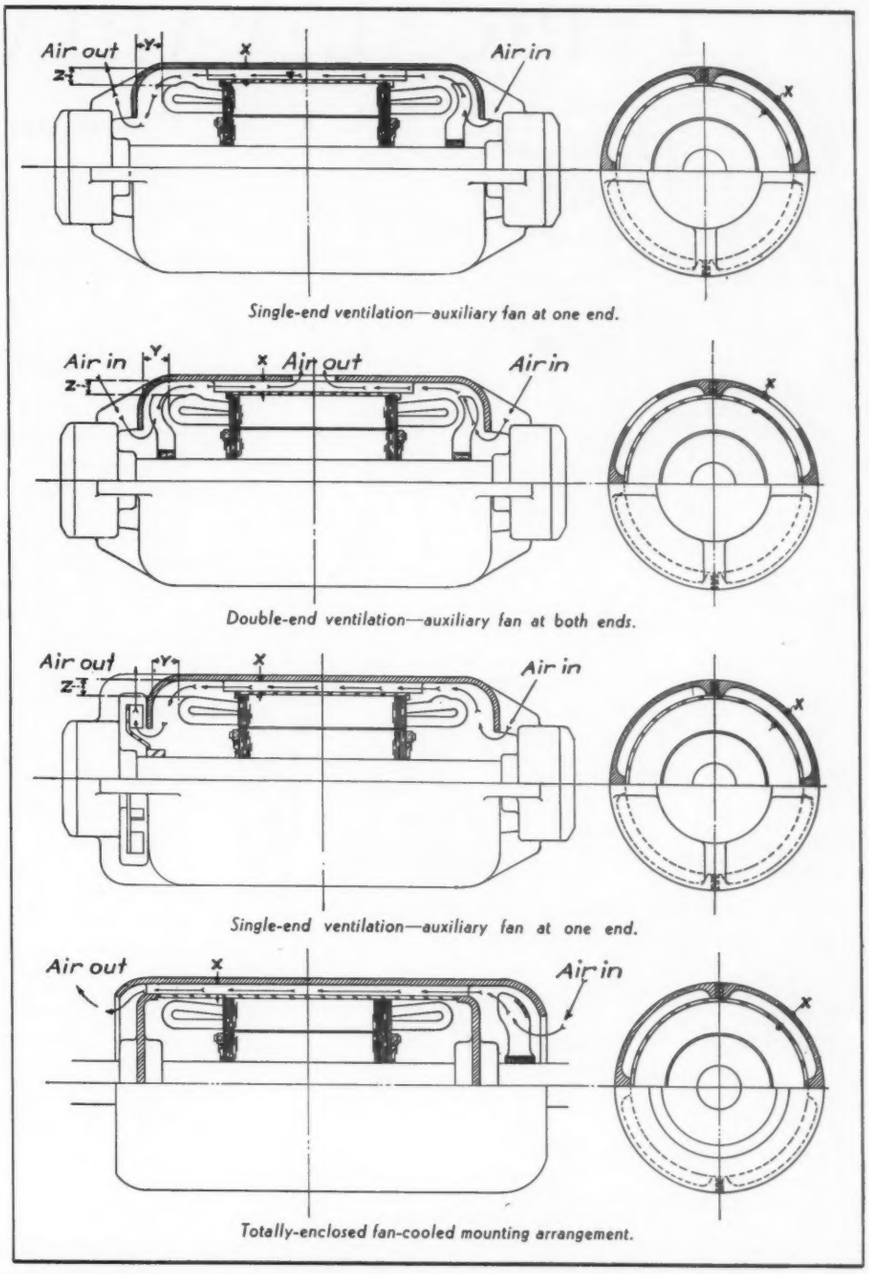


Fig. 2. Different methods of ventilation and various kinds of housing. Note type and relative diameter of fans. See table below for dimensions

TABLE III. Typical Mounting and Ventilation
Rolled Shell Shaftless Motors

Values of X, Y and Z refer to dimensions shown in the figure

Stator Dia. (in.)	*Total Duct Area (sq. in.) "X"*	*Min. Clearance (in.)* "Y"	"Z"	**Fan Blade Width (in.) (Approx.)*
4.703	5	½	⅜	¾–1
5.703	6	⅝	½	⅞–1⅛
6.468	14	¾	⅝	⅞–2
8.000	19	1	¾	⅞–2
10.000	25	1	¾	1⅛–2
12.375	30	1½	1	1½–2¼

*Approximate fan blade width if fan diameter is same as external diameter of rotor based on using 8 to 12 blades. Width can vary between these limits directly with the length and horsepower rating of motor. Above applies to speeds of 3600 rpm. and above. Lower speed ratings require fans having a diameter approximately equal to diameter of stator.

TYPICAL VARIABLE-SPEED TR

Mechanisms employing one or more expanding pulleys for variable speed driving of heavy or light loads covering various ranges of speed ratio, adjustable during operation

CYRIL DONALDSON
Rochester Athanaeum & Mechanics Institute

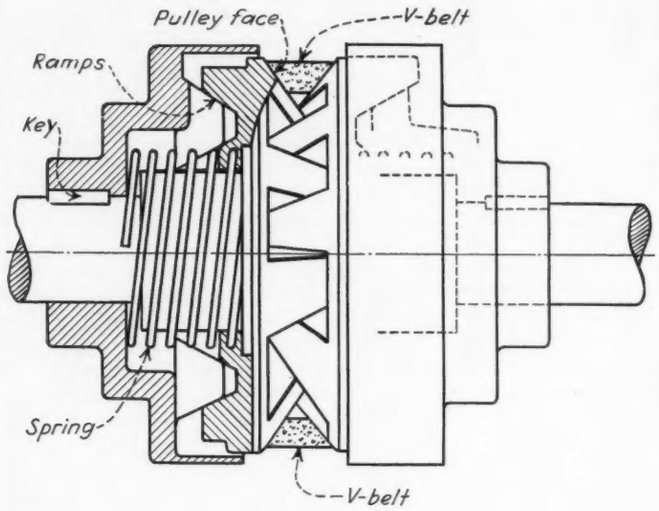


FIG. 20

FIG. 20—In this "Hi-Lo" driver pulley, the telescoping faces turn and slide freely on their spindles. No splines are used, but the sheaves are caused to rotate with their respective spindles by means of mating ramps (or cams) on hub and sheave. Belt rides in vee of sheave at desired diameter, and tension is automatically adjusted by the pressure of the spring. Speed ratio is changed by sliding motor longitudinally on its base by means of power screw, causing belt to climb or sink in sheave vee. Driven pulley is of the plain vee type.

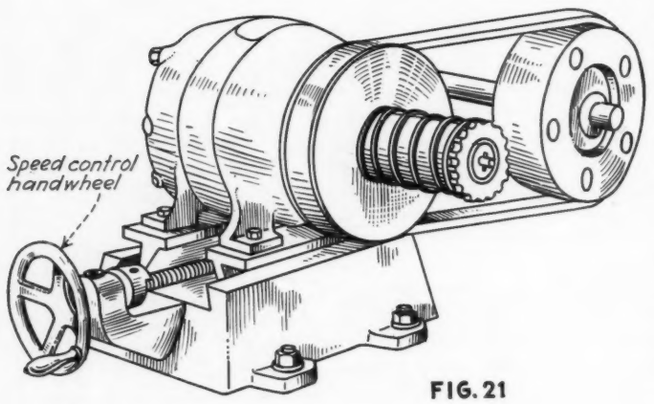


FIG. 21

FIG. 21—Similar to Fig. 20, this Reeves Vari-speed motor pulley has the inner sheave fixed to the shaft and the outer sliding on a spline. Spring acts against the sliding sheave in order to provide sufficient pressure against the sides of the belt.

FIG. 22—Using two vee belts and an intermediate countershaft, this Columbia Vari-speed J.F.S. Jr. transmission, in order to vary the speed ratio, causes one belt to sink in one countershaft pair of sheaves while the other rises. Uniform belt tension is maintained for all speeds by rocking counter shaft. Desired radius of belts obtained by sliding inner countershaft endwise. No splines or springs are used.

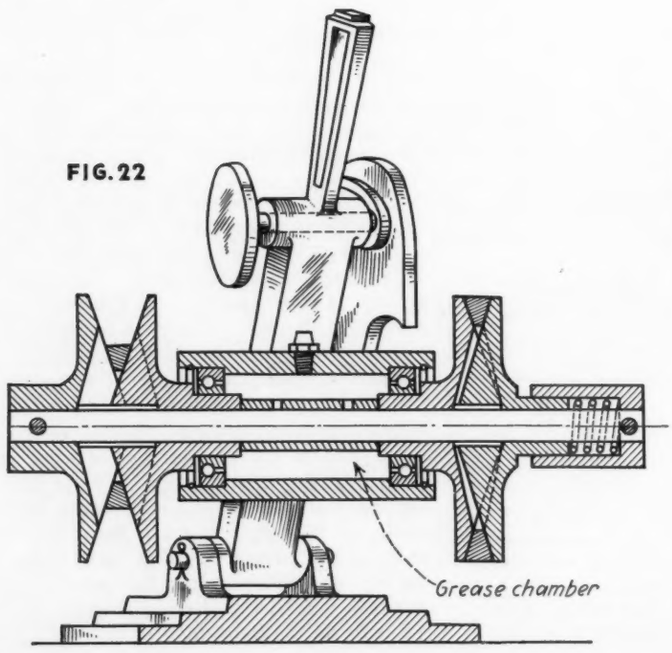


FIG. 22

FIG. 23—Similar to Fig. 22 this Speedmaster unit employs three belts and a centrally pivoted rocker arm. Velocity ratio changed by rocking arm which varies effective diameters of pulleys and keeps bolts at proper tension.

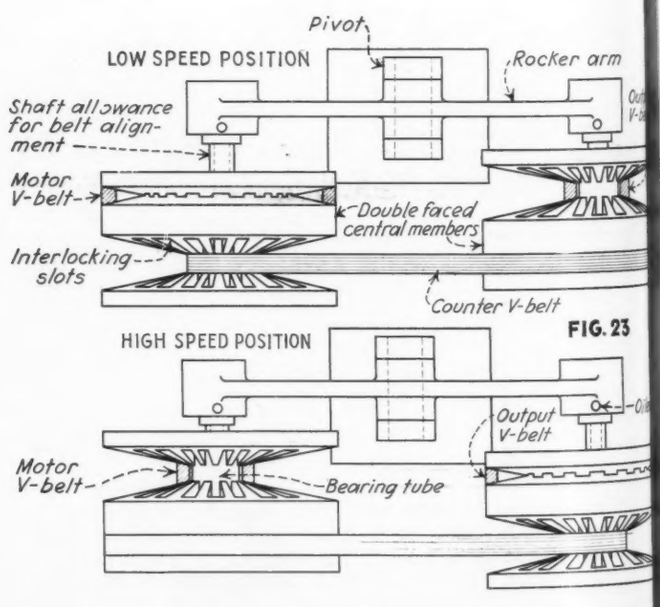


FIG. 23

TRANSMISSIONS–III

FIG. 25—In this well-known Reeves transmission, both pulleys are of the expanding type. Correct tension of the wide vee belt at any given speed ratio is maintained by tying the four sliding disks together with a linkage system. Can be had with built-in motor or as a separate unit. Belt can be of special wood-link type or of corrugated rubber. Speed ratio changed by right and left hand screw actuating levers so as to bring together one pair of disks while the other pair is simultaneously separated, thus varying the effective diameters.

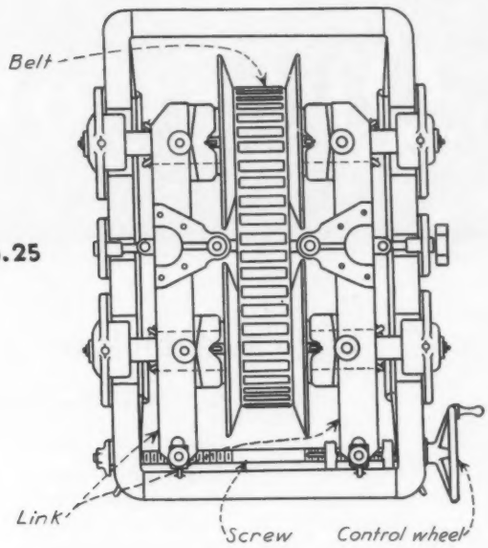


FIG. 25

FIG. 26—Similar to Fig. 25, this Lewellen transmission features the wide use of ball bearings.

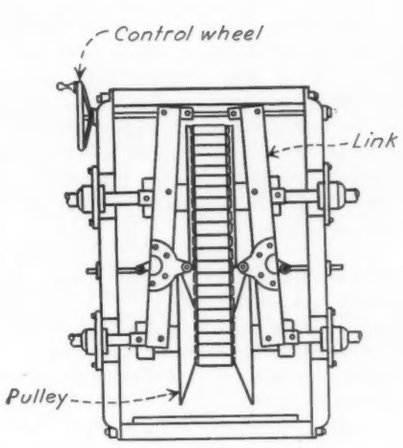


FIG. 26

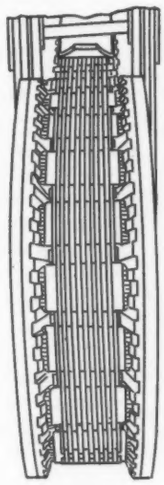


FIG. 28

FIG. 27—This unit is similar to Fig. 26, but has only two levers in the linkage system and a coiled spring for maintaining the correct belt tension.

FIG. 28—In this P.I.V. transmission, positive drive is ensured by using a chain as the flexible connector between the two shafts. When the speed ratio is changed by endwise motion of the sheaves causing the chain to raise or lower, chain laminations slide laterally in order to adjust themselves to tapering slots in the sheaves. Linkage system similar to Fig. 25.

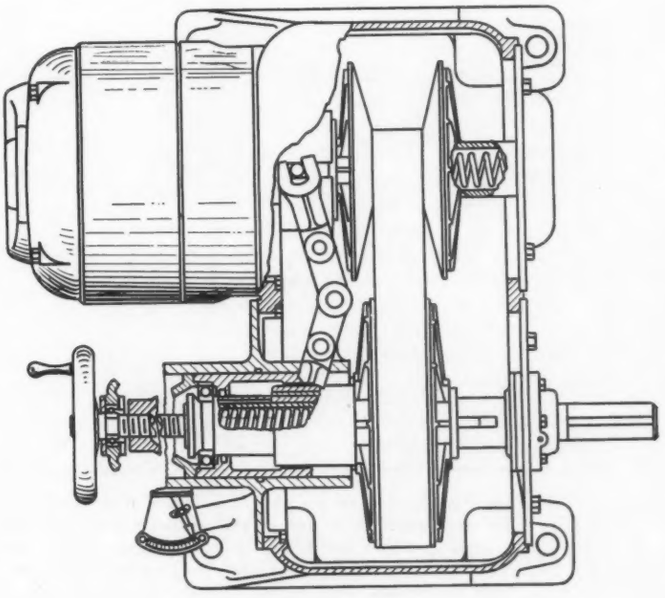


FIG. 27

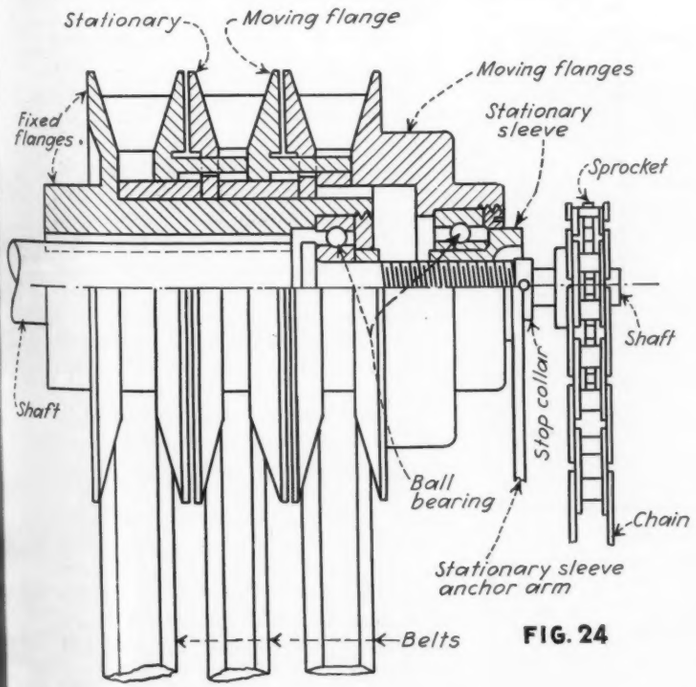


FIG. 24

Question and Comment

Logical Attack on a Packing Problem

N. V. Smith *President Smith-Johnson Corp.*

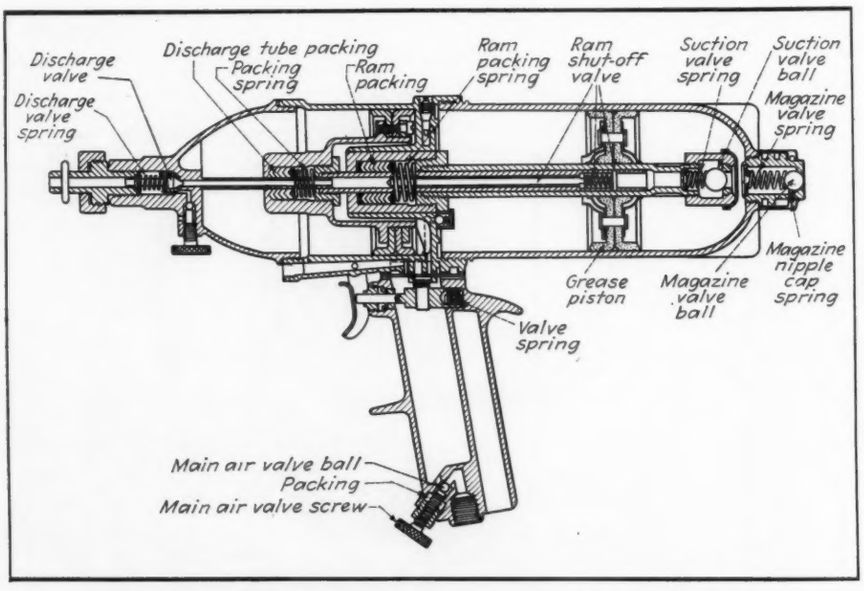
Manufacture of high pressure lubricators designed on the same principle as displacement type pumps must have some type of ram packing capable of withstanding the high pressures generated. Delivery pressures of these pumps vary from one hundred to ten thousand pounds per square inch, with a material which by its very nature is difficult to pack off. We have used, and probably always will use, this basic form of construction for reasons which are outside the scope of this article, hence we were early faced with the necessity of looking at the "packing point" with more than usual care.

Our earliest unit employed the outside packed form of construction which, given expert maintenance, is probably the best. In lubrication equipment, however, maintenance is often conspicuous by its absence, therefore we have had to develop a type of packing which would require no maintenance for long periods of time. We have accumulated considerable date on packing problems through this development work.

Most handbooks on engineering design show forms of packing boxes which indicate that the designs for sealing off a fluid substance around a moving member are the results of conjecture rather than logical attack. We found early in the game that no help could be expected from the packing material salesmen because they were not familiar with our requirements. We therefore set down the design points:

First, we were attempting to pack off a material, lubricant, which is designed to cling to and thoroughly wet the surfaces it is applied to.

Second, some form of automatic take-up of wear of the packing material was highly desirable, assuming that maintenance could not be counted on.



Sectional view of pressure lubricator with packing box designed by systematic procedure. On pressure tests the rams have been forced to make upward of a half million strokes against six thousand pounds delivery pressure without any apparent leakage. In this apparatus, this would indicate a life of from two to three years before repacking

Third, long life between repacking was desired to prevent condemnation of the product because of too frequent servicing.

Fourth, design so that repacking would require complete renewal of packing material and eliminate estimation of the amount of such material needed.

Fifth, the packing should be efficient for a period of time close to its extreme life so that small, annoying leaks would not occur.

Sixth, Design for use with materials having a wide range of viscosity and "wetting" ability,—from oils to heavy fibrous greases—and for performance under weather conditions varying from sub-zero to desert heat.

Attacking our first point brought out the fact that we could not hope to wipe the ram clean by mere contact, as a lubricant stands up under much more severe wiping than could be given it in a packing gland. We then tried a conventional packing with a closely wound spiral bronze spring around the ram to break the oil film. This worked well temporarily, but the spring, which was analogous to a piston ring, soon wore out. Hence, we discarded that method.

By a little experimenting we found that for wiping a greasy rod by hand with various materials, an absorbent substance gave the best results. Then, thinking back to our school day experiments, we remembered that animal tissues, such as skins or leather, not only have an absorbent effect, but also develop osmotic pressure of the absorbed substance in one direction through the skin. Flat washers of leather were tried in the packing gland, with the hair side toward the pressure, and were very successful; but they did not satisfy point two for, when they wore to a loose fit on the ram, absorption stopped. Also, life wasn't very long.

We now felt we were on the right track, however, and continued to work with leather, trying next a cup leather, and a series of cup leathers. They lacked sufficient mechanical strength, and did not always have enough initial seal to expand under pressure. When we increased the height of the cup lip to get this initial seal, the frictional drag was too great. Another fault of cup

leathers was the difficulty of holding them firmly without over-tightening. This defeated point three.

Backing up the packing on the pressure side by means of a heavy spring, and at the same time using a chevron style leather ring, six in each box, satisfied point two. The chevron style was selected and used with the back-up spring in an attempt to get a good initial seal without too much drag. These points were achieved beyond expectations, and we then proceeded to test this design against the other points.

Point one was satisfied by using leather with just enough wax in it to hold the shape of the formed rings until used, with a wax that was soluble in the lubricant pumped,—that is paraffine and oil. Thus, in use, we would have a material that was absorbent and would wipe clean.

The second point was satisfied by the spring take-up, as mentioned previously.

The third point was satisfied because the leather, being in 90 deg. chevron form, would stand a lot of edge wear before becoming a flat disc.

The fourth point was satisfied—when the V-form of the packing was lost, the whole set of packing went bad at once, and since the packing box was equipped with a spring take-up, it was merely necessary to count out and insert six new rings.

The fifth point was also covered by the above changes.

The sixth point was satisfied by the selection of leather—a packing material which was practically constant in its characteristics over the range of weather conditions encountered.

We have detailed the steps taken to show that nearly every such enginering problem, if studied in a logical manner, can be solved and, at the same time, will provide a fund of future knowledge for similar problems. The writer does not wish to say by this article that leather is the panacea of all packing problems—it isn't—but if design engineers will not wait until the completion of their designs before tackling the packing problem, they will save a lot of time and money.

Tube End Reinforcements

To the Editor:

It seemed to me that I should call your attention to a couple of errors in the writeup of the Hart Process, page 138 of the April number of *Product Engineering*. In the first sentence you mention that the reinforcements are used for threading, welding, and other purposes. The only purpose of this reinforcement is in connection with a special process of welding; it would be of no use whatsoever in threading. You mention that it is covered by U. S. Patent 2,091,082. This is actually Patent No. 2,091,982. Actually the process is a method of welding two tubes, or a tube and another article, together, and the reinforcing is used merely to provide a heavier end for the welding, and also to distribute stress away from the actual weld.

—I. WHITEHOUSE
Steel and Tubes, Inc.

Drafting Shortcuts

To the Editor:

In the article "Drafting Room Practices," published in March and April, some suggested simplifications to effect economies are so hair-splitting they would give an intelligent draftsman the jitters.

For example, a pointer line indicating a radius on the end of a shaft has a whole arrowhead, but a pointer line indicating a chamfer at the same point on the same piece is allowed only half an arrowhead. I can just imagine a real draftsman trying to control himself when the checker returns a drawing to change whole arrowheads to half arrowheads. Also, it is simpler and clearer to show a chamfer and indicate its size than to show the end square and print out the word "chamfer" with no specified size.

As most drawings are made with pencil, there is nothing to gain by drawing a slot with one heavy line instead of two ordinary lines because two strokes of the pencil are required to make a heavy line. It takes no more time to draw two lines a short distance apart.

Any draftsman worth his salt does not waste his time on unnecessary motions. Those points should be taught the draftsman during his early training period. Rigid rules about petty details will never make a good draftsman any better.

—C. A. JOHNSON
Philadelphia, Pa.

Definition of a "Slug"

To the Editor:

I have just returned home after a long absence and have just read the article on "Viscosity—Analysis and Significance of Common Units" on page 353 of the September, 1937, number of *Product Engineering*. It is a very good article and I should like to see more articles of this type in your magazine.

However, there is one rather obvious mistake. The author gives the wrong definition of the unit called a "slug." The correct definition is given in *Mark's Handbook*, page 73, along with the misinformation that it is never used in practice. For a discussion of this unit and its meaning, I refer you to "*Technical Mechanics*" by Maurer and Roark.

I believe you have never referred to this error in your magazine. In our present-day treatment of flow problems and heat transfer, and other phases of engineering, the conception of viscosity is important. This article goes a long way to help most of us use modern data and formulas more intelligently. May I suggest more articles on this subject?

REUBEN A. WILSON
Milwaukee, Wis.

To the Editor:

Mr. Reuben A. Wilson is correct and I was wrong in defining the "slug" in what might be called the preamble to my article on viscosity in your September 1937 number.

The "slug" should have been 32.15 lb.-mass and not 1 lb.-mass as stated. I sincerely regret my error and I hope those who noticed my article will also notice this correction.

In Maurer & Roark's "*Technical Mechanics*", the authority mentioned by Mr. Wilson, there appears on page 194 the following statement regarding the "slug":

"It is a relatively new term (slug) and is not in general use. It was proposed by Professor A. M. Worthington in his '*Dynamics of Rotation*'. He took it from the word 'sluggish'. Other names have been proposed from time to time, for example; 'matt' from matter, 'ert' from inertia, 'gee-pound' from 32.2 lb., etc. But only 'slug' has any vogue; it is used freely in some literature on aerodynamics as in reports of advisory committees for aeronautics of Great Britain and of the United States."

In our New York Public Library, I was able to consult a second edition of "*Dynamics of Rotation*" by Professor Arthur M. Worthington, M.A., who was professor of physics at the Royal Naval Engineering College, Davenport, England. He lived from 1852 to 1917. Reference was made to unit of mass being 32.2 lb.-mass in the English gravitational system, where the three fundamental units are the foot, the pound-force and the second. However, I saw no mention of the "slug"; it may have appeared in later editions.

I am grateful to Mr. Wilson for his correction, for if ever I run across a "slug" again, it will have no terrors!

—W. F. OWEN
New York, N. Y.

How Many Teeth in Contact?

To the Editor:

Standard practice, as recommended by the British Standards Institution, and followed by at least the important gear concerns, would answer J.J.G.'s question, on page 120, March number

of *Product Engineering*, by saying that normal calculations should not be based upon more than one-tooth contact. On the other hand, it would regard the assumption of the total load as acting at the tip of a tooth as being rarely justified, since at the moment when the tip makes first contact another tooth is bearing all the load, and not until the line of contact has passed some distance down the profile is the entire load borne by a single tooth.

Hence in the British standard horsepower formula the strength factor *Y*, corresponding to the Lewis outline factor *Y*, has a value nearly twice that of the Lewis. In the given example, for instance, where the strength of the 24-tooth gear is equivalent to that of a 34-tooth gear, the B.S.I. factor *Y* for teeth of 20-deg. pressure angle, (the 14½-deg. tooth is no longer recommended), is 0.7 against the Lewis 0.371. The Lewis factor 0.327, applicable to 14½-deg. teeth, would become, say, 0.6, if modified in like ratio.

Your correspondent does not indicate the material proposed, but calculation places it as having a safe static stress of 14,000 lb. per sq. in. By the B.S.I. formula the horsepower works out as 0.93, after *Y* has been reduced, as mentioned above, to fit it for the smaller pressure angle. The new figure is so close to 1 as to justify our saying that 1 hp. would be transmitted by the gears.

American readers may be interested in another difference in the factor *Y*. In the Lewis formula *Y* is constant for each number of teeth; in the B.S.I. formula the number of teeth in the mating gear is taken into consideration. For instance, the factor for a 100/100 pair of gears is 0.8, whereas for a 100/20 pair it is 0.7, the factor being for the 100-tooth gear. Thus, when mating with a smaller gear, the 100-tooth gear is weaker than when mating with a larger gear.

But when we compare the relative strengths of each gear as a pair, the difference between the formulas is somewhat startling. Consider, for instance, the pair 24/120. The factor for the 24-tooth gear is 0.73; for the 120 gear, 0.72. Now consider a pair of the same ratio, but with fewer teeth, say 12/60. Now the factor for the smaller gear is 0.62, and for the larger gear 0.59. From the "factor" point of view, the larger gear is the weaker! This seems anomalous, but there is little reason to mistrust the findings of the committee of experts entrusted with the work of standardization, particularly as standardization was based upon the life's experience of the largest and oldest gear concern in this country.

Just a thought in conclusion: In view of the contradictory verdicts of equally competent instances, why not ignore the *Y* factor and always divide the calculated tangential load either by 2 to accord with the average B.S.I. factor or by 3 to accord with the average Lewis factor? Or . . . might we compromise at 2.5, the mean of these? With so sharp a divergence of opinion, it certainly seems ridiculous to continue to waste time in manipulating figures expressed to the third decimal place.

—Francis W. Shaw
Rochdale, England

To the Editor:

On page 120, March number of *Product Engineering*, J. J. G. asks:—"*How Many Teeth in Contact*" for a pair of miter gears of 3 Diametral Pitch, 24 Teeth.

J. J. G. is in error in assuming two teeth in contact and thereby expecting to increase the hp. output delivered. Assuming that two teeth were in contact, the original tooth load for one tooth as calculated would then be taken by two teeth thereby cutting down the original tooth load one-half, and not doubling up as he shows in order to gain his *one* horsepower.

Further, he has taken the pitch line speed in f.p.m. from the largest pitch diameter of the gears, whereas, he should have taken a mean pitch diameter instead. The corrected calculation would then be as follows:—Lewis Formula being used.

Miter Gears, 3 d.p., 24 tooth, 2″ face, material cast iron. 5 r.pm.; largest pitch diameter equals 8″; smallest pitch diameter equals 5.172″; mean pitch diameter equals 6.586″; mean diametral pitch equals 3.644″; pressure angle equals 14½ degrees; r.p.m. equals 5; factor *Y* Lewis formula equals 0.3.

Then V equals 0.262 × 6.586″ × 5 equals 8.628 f.p.m.

$$S = 8000\left[\frac{600}{600+8.628}\right] = 7888 \text{ lb.}$$

$$W = \frac{7888 \times 2'' \times 0.3}{3.644} = \text{say,}$$

1300 lb. safe tooth load.

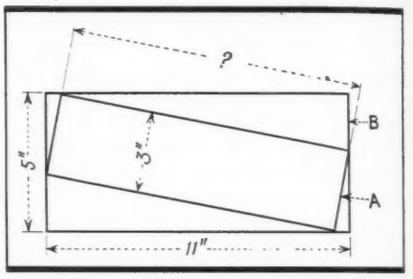
$$\text{Hp.} = \frac{1300 \times 8.628}{33{,}000} = 0.339 \text{ hp.}$$

Note:—Calculation actually shows that nothing less than a miter gear of 2 diametral pitch, 24 teeth, 3″ face, 12″ largest pitch diameter, 5 r.p.m., material being cast iron, can be used to deliver one horsepower. —Edward J. Rantsch
Hollis, N. Y.

Drafting Problem

To the Editor—

Recently a problem arose in our shop of putting the rectangular block *A*, across the rectangular block *B*, approaching the diagonal. The problem seemed quite simple until our draftsman discovered that he could not find



a way to lay it out correctly on the board. The next jolt came when he attempted to compute the length. What is the method of procedure for making a line drawing of this figure to scale?

—C. W. Hinman
Chicago, Ill.

Can You Work This One?

H. E. SMITH

Solution to Last Month's Problem—

Ship and Its Boiler

Let *X* = present age of ship and *Y* = present age of boiler

Then, in *Z* years, the ship's age will equal $X + Z$ and the boiler's age, equal to $Y + Z$, will equal $2X$, by the conditions of the problem.

Also, $X + Y + 2Z = 51$, and by substitution $5X - Y = 51$.

In order to satisfy the first conditions of the problem, *X* must equal 12 and *Y* must equal 9.

This Month's Problem—

Ladders in an Alley

Two ladders, 30 and 40 ft. in length respectively, are standing in an alleyway between two tall buildings. The base of each ladder rests against the edge of a building, and the top of the ladder leans against the building on the opposite side. The base of the one ladder is on the left side of the alley and the other on the right, so that they cross each other at a point 10 ft. from the ground. How wide is the alley?

News

Westinghouse Installs Creep Test Machines

FINISHING touches have just been added to the third of a battery of new electric furnaces by means of which engineers at the Westinghouse Research Laboratories will study the creep of steel and metal alloys. The three new testing machines were designed by P. G. McVetty.

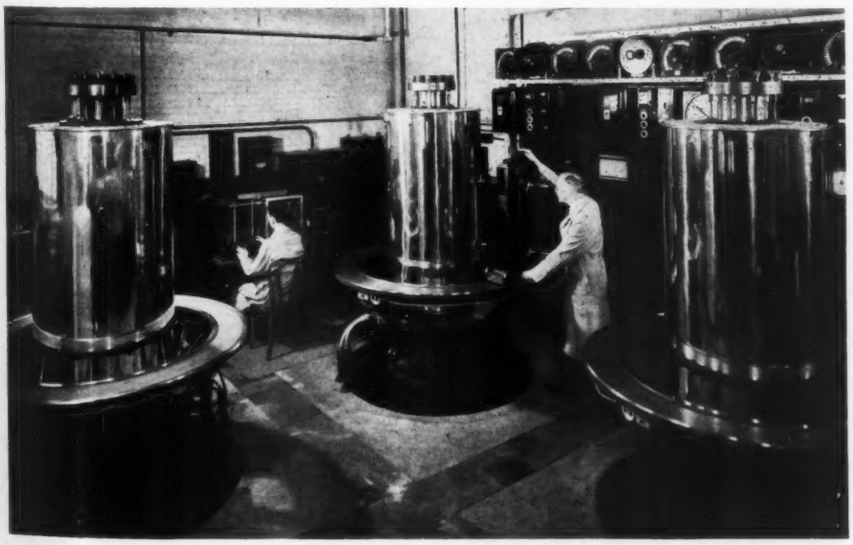
The design, in effect, has combined six creep machines into a single unit. With all three of the new machines in operation, it will be possible to conduct 180 simultaneous tests. Each machine is a heavy alloy steel block housed in a three-walled cylinder and supported on a foundation of sand. To retain the heat inside the furnace, the outer shell is made of concentric sheets of polished nickel and aluminum separated by powdered silocel. Sand was used in the 6 ft. foundation pit because of its ability to minimize the effect of vibrations which are known to hasten the creep of metal. To keep the sand warm and dry, the pit is equipped with a separate electrical heating unit.

Three electric windings on the metal core of the furnace produce temperatures up to 1000 deg. F. and a photoelectric cell maintains the temperature within 10 deg. by automatically operating a resistance which controls the electric current. The cylinder revolves once an hour in order to distribute the heat equally to all parts of the furnace.

When all three units are in operation, the operator can plug in by means of a telephone switchboard and determine how the heat is distributed inside the furnace. One hundred and twenty-five pairs of wires connect thermocouples in the furnaces with remote instruments for measuring and recording the temperature.

Each heating core has 12 spaces for holding 12 test samples 20-in. long which may be subdivided into five sections to make 60 tests. Dial gages connected with comparison rods extending through the top of the furnace measure the relative vertical displacement of the rods by samples which may be loaded with weights and levers to carry 50,000 lb. per sq.in. of metal under test.

For a double check, the machine is equipped with a circular track and a micrometer microscope type of extensometer. Two platinum spots or targets welded at the top and bottom of the sample metal, and scratched with very fine lines, are fastened in the furnace. For measurement of the creep, the cylindrical shell is revolved until the sample is opposite two quartz windows in the 10½ in. wall. A microscope, rolled in place on the track, sights through the quartz windows and enables measurement of the distance between the two platinum targets within 1/100,000th of an inch. Readings are repeated daily for approximately three months, and the results charted.



Newly completed creep test machines at the Westinghouse Research Laboratories can simultaneously test sixty samples

New Lab for Testing Finishes

RESEARCH LABORATORY devoted to the study of the principles of corrosion and the development of new processes for corrosion prevention and control, and a testing laboratory devoted to studying the comparative value of rust proofing processes and methods of corrosion prevention have recently been completed by Parker Rust-Proof.

In connection with the laboratories, a technical library is maintained containing all the more important publications dealing with chemistry, especially that phase of it pertaining to the chemical treatment of various metals for the prevention of corrosion.

The testing laboratory is equipped with processing tanks of such size that processing can be done under normal production conditions. Supplementing this, paint-spray equipment and electric baking ovens enable duplication of any type finish for test purposes.

Ford Casts Steel Gears Centrifugally

SPINNING molten steel at high speed to form cast steel parts of improved quality is one of the recent experimental developments of the Ford Motor Company.

"The centrifugal casting experiment has been confined so far largely to blanks for transmission cluster gears and ring gears," R. H. McCarroll, Ford metallurgical engineer, said in describing the development. "In most cases it has been entirely successful, resulting in stronger, lighter and less expensive gears than are produced by the conventional forging method. Centrifuging insures a sound casting, free of blow holes and shrinks, while any remaining crystal structure in the metal runs perpendicular to the forces the gear teeth will be subjected to in actual use."

It is claimed that it is possible to hold the dimensions much closer than in the case of forgings, leaving considerably less excess metal to be machined off the gear blank. The only sand cores required are those used where undercuts are desired for reasons of design, and to take care of shrinkage between prominent protruding parts of castings. At first the molds were spun at speeds

ranging from 400 r.p.m. to as high as 1,800 r.p.m., but 600 r.p.m. for the ring gear blanks and 800 r.p.m. for the transmission gear blanks have proved most satisfactory.

Ball Bearing Standards in One Volume

THREE standards for ball bearings approved by the American Standards Association, 29 W. 39th St., New York, N. Y., have been published in a single pamphlet. They are "American Standard Annular Ball Bearings—Single Row Type and Separable Type;" "American Recommended Practice Annular Ball and Roller Bearings—Wide Type;" "American Standard Angular Contact Type Ball Bearings." Copies of the new pamphlet are available at forty cents each.

Detroit-Harvester Acquires Prestole, Inc.

DETROIT-HARVESTER CO., manufacturers of automotive plant and farm implements has acquired the business of Prestole Devices, Inc., of Detroit, in manufacturing, marketing and licensing-to-manufacture sheet metal fastening devices for the automotive, household appliance and sheet-metal working industry.

Manufacturing operations are continuing at the Prestole plant in Detroit, with certain operations being transferred to Detroit-Harvester Co., Dura Division at Toledo.

Polaroid Used in Eye Testing Machine

TAKING ADVANTAGE of the properties of Polaroid, which polarizes light so that it vibrates in one plane only, a new eye test has been developed to eliminate fraudulent damage claims for impaired sight in either eye, according to the American Optical Company.

Mechanism employed consists of a Project-O-Chart for projecting test letters, and a single polarizing lens in front of the projector. This polarizer can be turned through a 90 deg. angle to change the polarization plane.

Two polarized lenses with the polarization planes at right angles to each other are placed before the subject's eyes. By turning the polarized lens in front of the projector from one point to another, the test letters become visible to the left or right eye of the subject at the will of the examiner, but totally unknown to the subject himself.

Method Measures 1/100,000,000 Inch

QUANTITATIVE measurement of single molecular layers of many substances is possible by a method devised in the General Electric Research Laboratories and reported in the May issue of the *G.E. Review* by Dr. Katharine B. Blodgett. Differences of thickness of one ten-millionth in. up to one one-hundred-millionth in. can be measured by the method.

Surface films of oil on water reflect bright patches and streaks of color. A streak of oil seven millionths of an inch thick has a straw-yellow color and a streak nine millionths of an inch thick appears red. The colored areas thus constitute a contour map of the topography of the oil film, the scale of contour levels from one color to the next being one or two millionths of an inch.

The method devised in the research laboratory employs sheets of barium stearate which have a uniform thickness of one-ten millionth of an inch. Each sheet consists of a single layer prepared by placing a small amount of stearic acid on water containing a barium salt. The stearic acid spreads out over the water until each molecule has a place on the surface. The molecular layers are transferred to a metal surface by a dipping process at a rate of about 20 layers a minute. Color gages are made by building on a polished chromium plate a series of steps having 21, 41, 61, 81, up to 201 layers of barium stearate. These steps have thicknesses of 2, 4, 6, 8 millionths of an inch, the range of thickness in which films reflect most vivid interference colors. Films of other substances such as iron oxide, showing interference colors, are measured by comparing the color of the oxide with the colors of the steps of the gage. By means of suitable optical apparatus, differences of one one-hundred millionths of an inch are measureable.

MEETINGS

AMERICAN SOCIETY OF MECHANICAL ENGINEERS—*Fall meeting, Providence, R. I. October 5-7, C. E. Davies, secretary, 29 W. 39th St., New York City.*

SOCIETY OF AUTOMOTIVE ENGINEERS—*National Transportation Engineering meeting, Commodore Hotel, New York, N. Y. Nov. 14-16, John A. C. Warner, secretary and general manager, 29 W. 39th St., New York City.*

EXPOSITIONS

NATIONAL METAL CONGRESS AND EXHIBITION—*Convention Hall, Detroit. Mich., under the auspices of the American Society for Metals, Oct. 17-21. Further information can be obtained by communicating with W. H. Eisenman, secretary, 7016 Euclid Ave., Cleveland, Ohio.*

NATIONAL AUTOMOBILE SHOW—*Grand Central Palace, New York City, Nov. 11-19, under the auspices of the Automobile Manufacturers Association, 366 Madison Ave., New York City.*

THIRTEENTH NATIONAL EXPOSITION OF POWER AND MECHANICAL ENGINEERING—*Grand Central Palace, New York City, December 5-10. Further details can be obtained from Charles F. Roth, president of the International Exposition Co., Grand Central Palace, New York City.*

Highway Reflectors of Molded Plastic

REFLECTORS, molded of Lucite, a flexible, non-shattering plastic recently perfected by du Pont de Nemours & Co., are being installed on an 85-mile stretch of main highway from Detroit to Lansing, Mich. This plastic is claimed to mold much more accurately than glass and to retain permanent transparency. The reflectors are set 100 ft. apart on either side of the road, closer on curves.

The markers made of Lucite are based on the optical principle known as retro-directive reflection. Of the two types of retro-directive reflectors, the one used is the triple reflector or cube corner. No silver or metallic reflecting surfaces are necessary. Efficiency of the markers is possible because of the accuracy with which the new plastic can be molded.

Technical Index

TECHNICAL contents of English language periodicals in the field of engineering, chemistry, physics, geology published throughout the world are now collected and published each month in a magazine called *Current Titles*. The contents are indexed both by subject and by the name of the periodicals included. Subscription price for *Current Titles*, 920 Broadway, New York, N. Y., is $3 a year.

Do You Know That—

ONE MILLION tons of steel are nickel plated each year in the United States and Canada? (82)

MACHINING PROCESSES have been developed for the manufacture of high precision miniature ball bearings of true radial type. Bearings are available in sizes down to approximately $\frac{1}{16}$ in. outside diameter. (83)

New Materials and Parts

Self-Contained Bearings

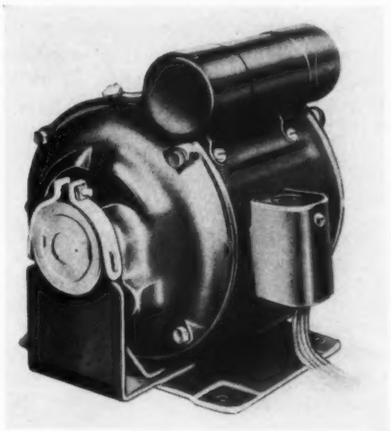
Bearings of the needle type which are self-contained within a roller race that is completely filled with the rollers and employs no roll separator. Construction is such that rollers of greater length than usual can be used for a given overall length of bearing. End flanges are square with the shaft. Lubrication is effectively retained and foreign matter excluded because of the close fit between the bearing and the shaft or spindle. Available for journals ranging from $\frac{3}{8}$ to $1\frac{1}{2}$ in. diam. for load ratings at 100 r.p.m. ranging from 791 to 16,000 lb. The Heim Co., Fairfield, Conn.

Timeter

An electrically operated counter which connects across the terminals of any electrical device or motor-driven machine. It is essentially a small, slow-speed, self-starting synchronous motor which drives a set of numbered wheels through an oil inclosed gear train. As it automatically starts and stops when the current supply to the device or machine is connected or disconnected the Timeter will indicate total hours of operation. Maximum registration is 99,999.9 hours, the figure indicating tenths of an hour being of a different color. Can also be supplied to read minutes and tenths of minutes if desired. The instrument is 3-9/16x3¼x2⅛ in., regularly supplied for surface mounting, although flush mounting can be obtained. Standard rating is 100-125 volts, 60 cycles, 2 watts. Other voltages and frequencies, including d.c., are available at a small additional cost. List price is $20. National Instrument Co., 44 School St., Boston, Mass.

Double-Capacitor Motor

Equipped with two condensers of different value—a high value of capacitance for starting and a low value of capacitance for running, the change in capacitance occurring at approximately 75 per cent full-load speed by means of a centrifugal switch on the motor shaft. The characteristics of these



high-torque double-value capacitor motors are similar to other double-value capacitor motors, giving high starting torque, low starting current, and high power factor and efficiencies. In appearance and construction the motor is similar to other Wagner capacitor and split-phase motors of the same size, although the capacitor box is slightly larger. Type RZN is equipped with rigid mountings and type RZNR is equipped with annular resilient mountings. Available in 110/220 volts in sizes 1/3, ½, ¾ and 1 hp. ratings. Wagner Electric Corp., 6400 Plymouth Ave., St. Louis, Mo.

Rectifier for Arcs

To provide d.c. of the proper characteristics for all makes of 6 millimeter carbon arc lamps, this rectifier, catalog

1072, is designed to convert 115 volts a.c. to 60 volts, 5 amp. d.c. Rectification is obtained through the use of a Raytheon RX-207 tube. A built-in current limiting transformer gives the same operating results formerly obtained with wasteful external ballast resistors. No adjustments are required. The unit plugs into the 115 volt a.c. supply and the arc lamp is plugged into the d.c. output receptacle of the unit. Raytheon Mfg. Co., 133 Willow St., Waltham, Mass.

Synthetic Rubber Lined Hose

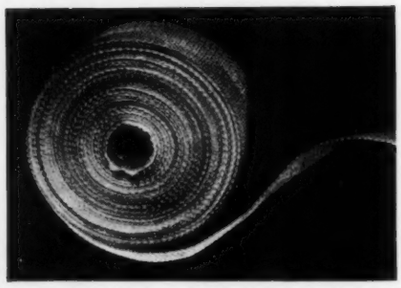
Air-drill hose, constructed with an inner lining of gas and oil-resistant synthetic rubber to prevent deterioration when oil gets into the hose. A heavy abrasion-resisting cover of black rubber gives the hose added resistance to abrasion and increased strength. Available in all regular sizes from ¼ to 2 in. The Manhattan Rubber Mfg. Div., Raybestos-Manhattan, Inc., Passaic, N. J.

Testing Instrument

Known as the S-P-S tester, for testing the printing quality of paper, including compression, softness, porosity and smoothness. The instrument measures a given air-leakage either over or through the paper. Air, at uniform low pressure, is supplied by an inverted cylindrical cup floating freely in an outside cylinder partly filled with oil. Graduations on the inner cylinder are timed as the cylinder displaces air and drops. Tests are made by clamping the sample between a fixed upper orifice plate and one of three interchangeable lower plates, as selected for testing softness, smoothness or porosity. W. & L. E. Gurley, Troy, N. Y.

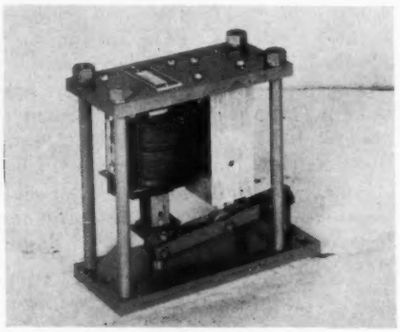
Glass Insulating Tapes

Intended for the insulation of coils for motors, generators, transformers, and for cables and other electrical conductors. Woven entirely from glass yarn, they not only have exceptional electrical properties and chemical resistance, but have the ability to withstand temperatures far in excess of the limits specified for Class B High-Temperature Insulation. They impregnate readily with resins, gums, and varnishes to form an insulation impervious to moisture and of high dielectric strength. Corning Glass Works, Fibre Products Div., Corning, N. Y.



Air Control Valve

A new control valve of the "Quick-As-Wink" line for the operation of air cylinders of diameters not over 3 in. with correspondingly short strokes. Available in $\frac{1}{8}$ in. I.P.S. only suitable for air pressure up to 200 lb. operating on G.E. 290-D solenoid with low amperage characteristics. Valve is being furnished for two-way and three-way operations, and can be supplied normally closed or normally opened. C. B. Hunt & Son, Salem, Ohio.

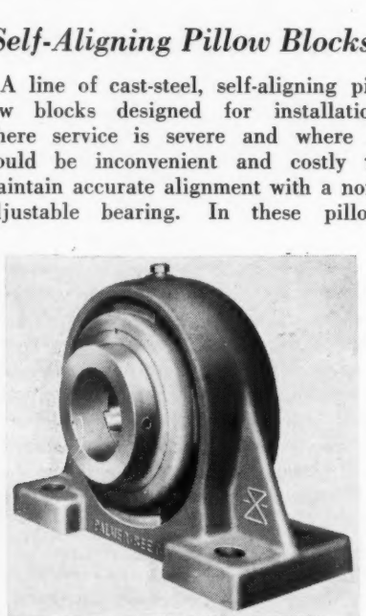


Welding Tip

Fabricated to give greater uniformity in strength, hardness and electrical and thermal conductivity, this new type spot-welding tip is intended to give longer life. Metal is uniform from welding face to water hole and length of this section was determined by tests to give maximum number of welds. Round bottom water holes leave no corners to trap steam. The company also announces completion of standardization of its line of welding tips and water-cooled holders. P. R. Mallory & Co., 3031 E. Washington St., Indianapolis, Ind.

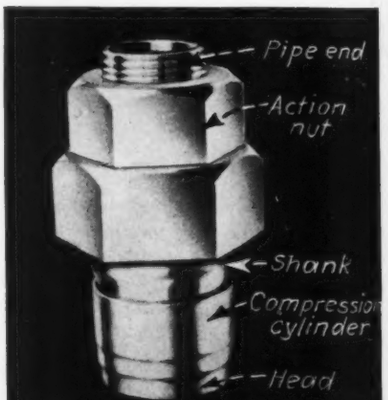
Self-Aligning Pillow Blocks

A line of cast-steel, self-aligning pillow blocks designed for installation where service is severe and where it would be inconvenient and costly to maintain accurate alignment with a non-adjustable bearing. In these pillow blocks the true ball and socket, or oscillating movement, is obtained by machining the bearing and supporting housing to a spherical surface, and accurately seating the bearing in the housing. This design permits a free oscillating movement in any direction of approximately 3 deg. max., and at the same time maintains an unbinding, accurate alignment of bearing and shaft. They are equipped with Hyatt heavy-duty precision roller bearings and require lubrication only at infrequent intervals. Available in shaft sizes from $1\frac{15}{16}$ to $6\frac{7}{16}$ in. Palmer-Bee Co., Detroit, Mich.



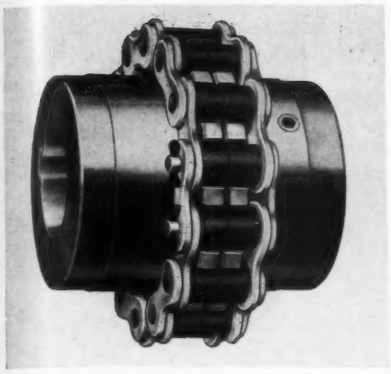
Multi-Seal Pipe Connector

An all-steel, multi-seal front drive pipe connector for securing pipes or fittings to all types of containers from the outside. The compression cylinder located between the head of the pipe and the connector shank moves up on the shank taper, expanding and compressing to the inner surfaces of the job as the smaller, or action nut, is tightened. The compression cylinder takes the convex curvature of the inside surface. The pipe end has a standard taper thread. The connection may be used on steam, gas, oil, water and air, and will withstand high pressures. Multi-Seal Mfg. Co., 123 N. Jefferson St., Chicago, Ill.



Flexible Couplings

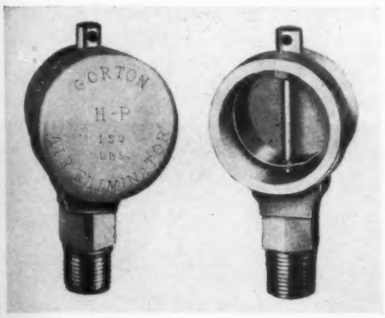
An improved coupling, Type RCB, consisting of two cut tooth sprocket wheels connected by a piece of specially constructed single-width finished steel roller chain, using a recently patented divided-roller feature which combines the advantage of double roller chain with the more rugged and simple construction of single width chain. The divided roller provides independent roller action for each sprocket, and as the contact between roller and sprocket causes the roller to revolve on its bushing, any tendency to scuff the rollers



and sprocket teeth is said to be avoided. Longer coupling life, and extension of the range of efficient application, are claimed for the new coupling. Link-Belt Co., 519 N. Holmes Ave., Indianapolis, Ind.

Automatic Air Eliminator

A high-pressure air eliminator for quick and automatic venting of dryers, steam coils, unit heaters, processing equipment, etc., under varying steam pressures up to 150 lb. per sq.in. The

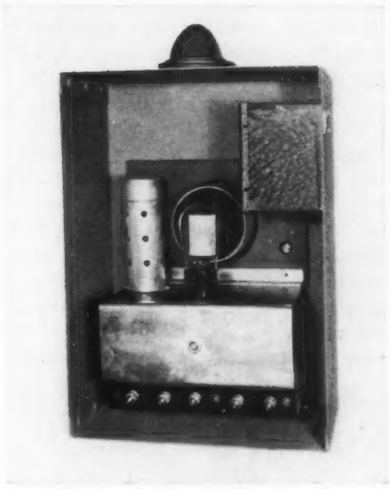


operating member of the unit is made of a special non-corrosive metal sensitive to heat changes. It closes the air eliminator at 212 deg. F. and keeps it closed until the temperature drops below that point, when it automatically opens the eliminator. The body of the air eliminator is made of bronze, nickel plated. It is 2½ in. in diam., 4⅝ in. high over-all, and has ½ in. male as well as ¼ in. female bottom connection. The size of the air outlet is $\frac{3}{16}$ in. diam. The unit must be mounted in an upright position. Gorton Heating Corp., Cranford, N. J.

Solenoid Control Relays

Two new solenoid control relays, designed to provide pump-free control without the danger of overheating or otherwise injuring the solenoid coil or closing rectifier. They are designated as Types PB-63 and PB-64, for moderate and heavy duty respectively. Except for contact ratings both types are similar. The solenoid closing coil circuit is de-energized at a definite point regardless of the control switch contacts or other maintained-closed contacts. Overheating of the solenoid coil is prevented even if the closing contacts of the control switch are kept too long in the closed position. Positive operation is assured, it is claimed, even for momentary closing of the contacts of the control switch. General Electric Co., Schenectady, N. Y.

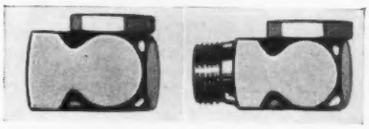
Photoelectric Counter Unit



A self-contained photoelectric counter control unit, using a standard mill type lamp. The unit has an ingenious light louver system to exclude all extraneous lights, and a highly corrected condensing lens to direct the full light energy upon the photocell. An amber light on top of the steel cabinet signals each step of the counter cylinder. The hinged cover has protection for locking the reset feature of the counter. Three-quarters of an inch separation of parts will serve to actuate the counter unit on 110 volts a.c. or d.c. supply. Wayne Automatic Relay Co., Fort Wayne, Ind.

Spray Nozzles

Designed to give a uniformly distributed hollow cone spray, highly atomized. These "Whirljet" non-clogging spray nozzles are of sturdy construction, accurately machined and available with removable cap in male or female pipe connection, ⅛ to ¾ in., and one-piece construction female pipe connection, 1 to 2½ in. Capacities range from 0.1 to 107 G.P.M. at 10 lb. pressure. All sizes and capacities are carried in stock made of brass and some sizes also in stainless steel. Spraying Systems Co., 4922 W. Grand Ave., Chicago, Ill.



Finishing Material

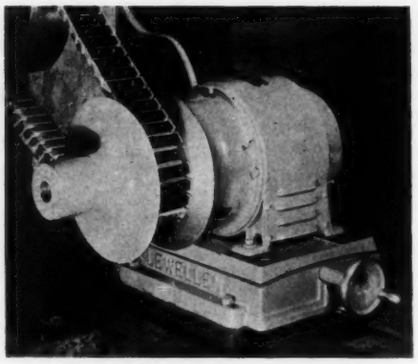
A new enamel for industrial finishing, known as Short-Bake "Dulux," which speeds up production by virtue of its shorter baking time, 8 to 10 min. at 350 deg. F., providing a good hiding with one coat. It may be applied on solvent cleaned steel, bonderited steel or primed steel and adheres well to all metals. It shows hardness, mar resistance, flexibility and high resistance to grease, soap and alkali. White or colors are available. E. I. duPont de Nemours & Co., Inc., Wilmington, Del.

Stiffness Gage

A precision-built instrument, known as the Taber V-5 stiffness gage, for accurate measurement of the stiffness and resilient qualities of paper, coated fabrics, light metallic sheet and wire. An important accessory to the gage is the triple-cut shear which blanks out test specimens accurate to size, insuring against dimensional errors associated with single shears. Both units are light and portable. Features of the gage include simplicity of operation; metric calibration with continuous scale from O to 2,000 gram-centimeters; no delicate parts, electrical contacts or connections; sensitive bearings sealed in lubricant for years of service without further attention; and low price. Taber Instrument Co., North Tonawanda, N. Y.

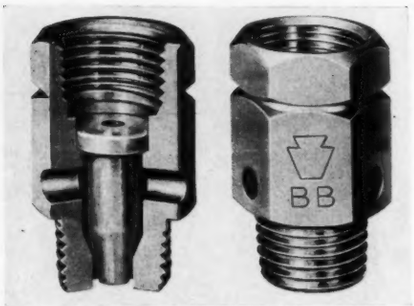
"Brawn" Monel Tubing

Made by a patented method of pressure brazing, then cold drawing, producing a homogenous tube wall which approximates seamless in tensile, compressive and bursting strengths. Manufactured from original strip to a very fine tube without the usual waste, labor and time, the tubing is produced at a considerable saving in cost. Available in sizes from $\frac{1}{16}$ in. O.D. (0.062 in.) x 0.007 in. wall, to ⅝ in. O.D. x 0.049 in. wall. Superior Tube Co., Norristown, Pa.



Motor Pulley

Variable-speed motor pulley which fits the shaft of a standard motor and drives directly to a pulley or sheave on the driven shaft. Motor and pulley mount upon an adjustable base permitting infinite variation of speed of the driven shaft. Special features of the pulley include positive equalized disk travel in both directions, maintaining belt in same position at all diameters; a uniform low spring pressure at all pulley diameters without buckling the belt; hardened steel driving members with ample bearing areas; pressure lubrication to all bearing surfaces with all working parts, including the spring, completely inclosed, keeping grease in and dirt out; high tensile castings with all surfaces machined, making a lightweight, safe, smooth-running drive. Lewellen Mfg. Co., Columbus, Ind.

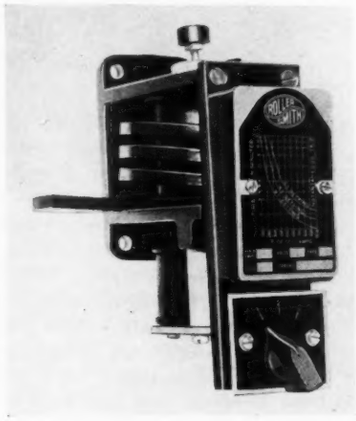


Safety Lubrication Control

Known as the Keystone BB fitting, used in connection with the grease cup or high-pressure fitting at the bearing to insure against uncontrolled lubrication. The maximum pressure under the control exercised by the device is 2½ oz. per sq. in. on the bearing proper, making it impossible to overload and force grease from the bearing housing into motor windings or other adjacent parts with resultant losses and damage to machine or product. The BB fitting operates on the principle of molecular friction, introducing the lubricant to the bearing through a central duct. When the pressure at the bearing reaches 2½ oz. back pressure forces the lubricant past a calibrated orifice in the base of the fitting and out through relief ports at the side. When excess grease appears at these relief ports it is a signal the bearing is sufficiently lubricated. Keystone Lubricating Co., Philadelphia, Pa.

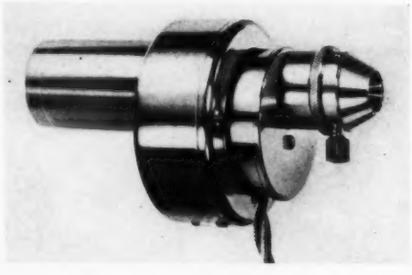
Thermal-Magnetic Trip

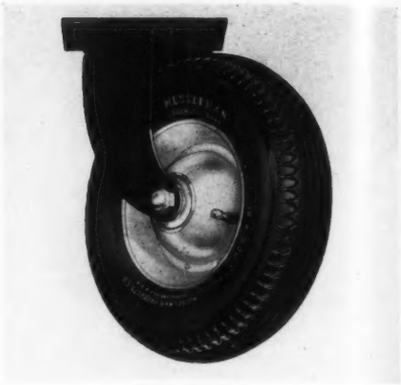
A new type of thermal-magnetic trip attachment known as Type DTA for the Types BD and HD air circuit breakers manufactured by this company. This thermal-magnetic trip attachment will permit high currents to flow for a safe period but will trip the a.c. circuit breaker before the dangerous temperature has been reached. If the current is high enough, as in a short circuit, tripping occurs instantaneously. It is recommended particularly for motor circuits to prevent tripping on starting currents. Available for currents from 600 to 6,000 amp. Roller-Smith Co., 233 Broadway, New York, N. Y.



Radiation Pyrometer

A new radiation pyrometer which uses as a temperature-sensitive unit, the Ardometer, as illustrated. This instrument measures the surface temperature of hot objects or masses above 1,000 deg. F. when the Ardometer unit is sighted so that it picks up the heat rays emitted. The temperature is recorded on a Bristol potentiometer, on a round or strip-chart recorder, or is indicated on a millivoltmeter pyrometer. The Bristol Co., Waterbury, Conn.





Castor Wheel

A new "Doenut" castor wheel with a tire 12 in. high and 3.30 in. in diam., and a total over-all height of 14 in. from floor to top of plate. The top plate is 7x4 in. and operates with a ring of ball bearings for easy turning. The plate contains four bolt holes for installation. The wheel itself is of a double disk type, cadmium plated, with adjustable ball bearings and equipped with a ⅝ in. cadmium-plated axle and acorn axle nuts. Measurement from top of plate to center of axle is 8 in. Engineered for loads up to 325 lb. per wheel. Musselman Products Co., 6308 St. Clair Ave., Cleveland, Ohio.

Pocket Dial Scale

No. 21 pocket dial scale that measures direct in 64ths, 32nds, 16ths, 8ths and larger fractions of the inch, with a full one inch measuring capacity. Moving the finger forward and backward against the operating wheel at the top causes the spindle and upper anvil to raise and lower and the dial to turn. The exact measurement of parts placed between the anvils appears in the dial slot. The scale is light in weight and fits the vest pocket. Case is polished and bright chromium plated. B. C. Ames Co., Waltham, Mass.

Speed Control Valve

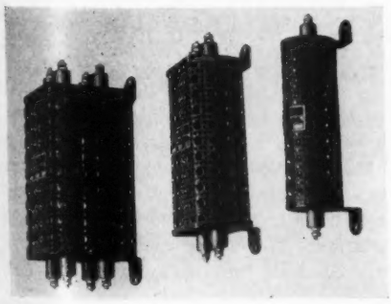
Designed to control the speed of an air cylinder at practically constant speed irrespective of manipulation of operating valve. This is accomplished by regulating the admission and exhaust of air to or from each end of the cylinder independently of the other end. Once set the valves require no further adjustment. Available in $\frac{3}{8}$, $\frac{1}{2}$, $\frac{3}{4}$ and 1 in. I.P.S. Hanna Engineering Works. 1765 Elston Ave., Chicago, Ill.

Corrosion-Resistant Coating

An addition to their line of corrosion-resistant coatings, Amercoat Rapid-Dry. It is a quick-drying solution which may be applied with a paint brush or with paint spray equipment to form a protective covering for exterior or interior surfaces that are subjected to the corrosive action of liquids, gases or solids. Claimed to resist the action of acid and alkali and other types of corrosive agents up to and including 20 per cent concentration. It is insoluble in salt water or in the petroleum derivatives. American Concrete & Steel Pipe Co., 4635 Firestone Blvd., South Gate, (Los Angeles Co.), Calif.

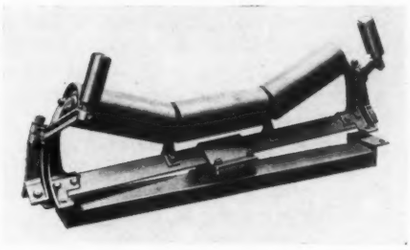
Cage-Type Resistors

A new standardized series of ventilated cage-type resistors available in three sizes—one, two, or four resistors. Suitable for use where it is desired to prevent accidental contact with the resistor. Can be mounted on switchboards and test panels, and in control, protective, or line voltage dropping circuits where wattages up to several hundred watts are to be dissipated. The cages consist of sturdy sheet metal ends with perforated metal sides finished with black wrinkle japan. Illustration shows cages for one, two or four resistors, 1 in. in diam. by $6\frac{5}{8}$ in. long. Overall dimensions are $2\frac{7}{8}$, $2\frac{29}{32}$, and $4\frac{1}{4}$ in., respectively, by $9\frac{9}{16}$ in. long. Mounting centers are $8\frac{3}{8}$ in. Cage resistors of different sizes and types are available from the manufacturer. Ohmite Mfg. Co., 4835 W. Flournoy St., Chicago, Ill.

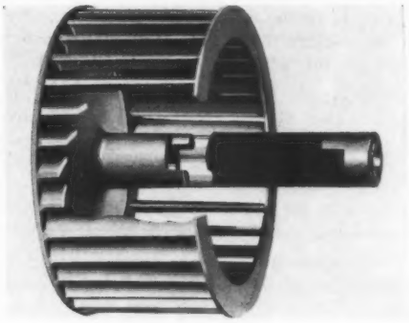


Self-Aligning Belt Idler

For automatically training conveyor belts without damaging the belt. Consists of a standard anti-friction idler pivotally mounted on a supporting cross member, with guide rolls at each end. The guide rolls are mounted on pivoted arms held in place by an easily removed locking pin. By swinging the arms through 180 deg. and putting the lock pin in place on the opposite side, the guides are made ready to train the belt when traveling in the reverse direction. As there is no pressure between the guide rolls and the belt there can be no damage to the edge of the belt. The guide roll moves outwardly the instant the belt touches it and in so doing swivels the idler sufficiently to cause the belt to return to normal position. The Jeffrey Mfg. Co., Columbus, Ohio.



Flexible Coupling



A new type "Double-Flex" flexible coupling, styled UX, for use on equipment where the load is not in excess of $\frac{1}{4}$ hp. Consists of two jaw units and a cushion unit made of solid, tube-shaped section composed of hard rubber, except where it contacts the jaws, where it is made of a softer, more resilient rubber. The L-R line of couplings are claimed to be noiseless as there is no metal-to-metal contact, give long life, and require no lubrication. They also automatically absorb shock, overload, torque and other stresses. Lovejoy Flexible Coupling Co., 5009 W. Lake St., Chicago, Ill.

Manufacturers' Publications

AIR ELIMINATOR—Gorton Heating Corp., Cranford, N. J. Bulletin 106, 4 pages, $8\frac{1}{2}$x11 in. Describing the Gorton high-pressure air eliminator for automatically venting unit heaters, dryers of various types, steam mains, processing equipment, etc., under varying steam pressures up to 150 lb.

AIRCRAFT BLUEBOOK—The Fafnir Bearing Co., New Britain, Conn. Aircraft Bluebook No. 2. Includes 36 pages of photographs and information on new aircraft types ranging from the smallest pursuit and passenger planes to the gigantic "flying fortresses." Typical Fafnir ball bearing installations in a variety of flight control systems, engine controls, and many other places in plane and engine constructions are shown.

BEARINGS—Chrysler Corp., Amplex Div., Box 2718, Detroit, Mich. Catalog, 32 pages, $8\frac{3}{4}$x$11\frac{1}{2}$ in. Catalog listing of their Oilite precision bronze bearings. Arranged according to sizes for plain, flanged and thrust bearings. Another folder describing Oilite bearings and a number of interesting engineering tests is also available.

BEARINGS — Norma-Hoffmann Bearings Corp., Stamford, Conn. Bulletin 962, 8 pages, $8\frac{1}{2}$x$10\frac{3}{4}$ in. Gives complete data as to sizes, dimensions and load ratings of precision needle roller bearings. Standard sizes of precision needle rollers are also given, together with engineering information. Includes drawings of typical applications.

CADALYTE—E. I. duPont de Nemours & Co., Inc., Electroplating Div., Wilmington, Del. Booklet, 12 pages, $8\frac{1}{2}$x11 in. A new technical service manual on Cadalyte "38" for cadmium plating. Recent improvements and changes in the product are discussed, together with operating instructions and methods of analyses. A table of costs and

time required for specified deposits is included. A new Cadalyte maintenance compound is also described.

COMPOSITE SILVER RIVETS—The H. A. Wilson Co., 105 Chestnut St., Newark, N. J. Folder 8½x11 in. General descriptive data, special features and advantages of Wilco composite silver rivets and a table of available stock sizes.

COOLING UNIT—American Instrument Co, 8010-8020 Georgia Ave., Silver Spring, Md. Bulletin 1085, 4 pages, 5½x8½ in. Details on their new portable cooling unit for use in baths or tanks where bath temperature is to be maintained at or below room temperature with an accuracy of plus or minus 0.1 deg. C.

DRAFTING ACCESSORIES—Engineering Sales Co., Two Rivers, Wis. Folder, 6 pages, 8½x11 in. Complete descriptive information, specifications and graphic illustrations covering Esco products manufactured by this company, including drafting equipment, accessories, and drawing board attachments.

DRAFTING MACHINE—Keuffel & Esser Co., Hoboken, N. J. Folder, 12x9 in. Information and numerous pictures on their Paragon drafting machine.

ELECTRONIC TUBES—Westinghouse Lamp Div., Westinghouse Electric & Mfg. Co., Bloomfield, N. J. Leaflet, Form S-860. A price list of Westinghouse electronic tubes for industrial applications.

FINISH—Parker Rust-Proof Co., Detroit, Mich., A.I.A. File No. 25-C-31, 16 pages, 8½x11 in. "A New Finishing System for Architectural Iron and Steel Products Increases Finish Durability," covering the advantages and applications of bonderizing.

FLEXIBLE SHAFTS—The S. S. White Dental Mfg. Co., Industrial Div., 10 East 40th St., New York, N. Y. Engineering Bulletin 38, 28 pages, 8½x11 in. Covering flexible shafts for remote controls, including descriptive data on shaft construction, specifications, torsional deflection charts and application procedure.

GEARS—Farrel-Birmingham Co., Inc., 344 Vulcan St., Buffalo, N. Y. Catalog 438, "Farrel-Sykes Gears and Gear Units," 80 pages. An engineering handbook containing complete information on the various types of speed reducing and speed increasing gear units and related products manufactured by the company. It is illustrated with pictures and diagrams and contains many comprehensive charts and tables.

G. E. BULLETINS—General Electric Co., Schenectady, N. Y. The following bulletins are available: Gear-Motors (1437C); Push-Button Stations (2473); Alnico Limit Switch (2907); Manual Motor-Starting Switch (2234B); Oilproof Push-Button and Selector Switch (2908); Combination Magnetic Switch (1587C); A-C Magnetic Switch, Size One (841H); A-c Magnetic Motor-Starting Switch (19H); New Track Type Limit Switch (2052A).

HOSE—Chicago Metal Hose Corp., Maywood, Ill. Folder, 4 pages, 8½x11 in. Descriptive data on their new Avioflex hose for oil connections to provide permanent resistance to action of oil at any operating temperature. Also includes information on Avio-Tite detachable couplings.

LACQUER—Hilo Varnish Corp., 42-60 Stewart Ave., Brooklyn, N. Y. Bulletin Sheet No. 23-L, 5x7⅜ in. Data on Hilo Trans-Lux metallic colored lacquer with samples.

LUBRICATORS—American Chain & Cable Co., Inc., York, Pa. Folder, 8½x10½ in. "Pressure Lubrication Without Special Fittings" illustrating and describing the Acco-Morrow pressure lubricator recently announced by the company.

MAGNETS—Stearns Magnetic Mfg. Co., Milwaukee, Wis. Bulletin No. 25. Featuring their line of circular, rectangular and hand magnets.

METALLOGRAPHIC EQUIPMENT—Bausch & Lomb Optical Co., Rochester, N. Y. Catalog E-225, 32 pages, 8½x11 in. Complete descriptive information and general specifications on Bausch & Lomb metallographic equipment and accessories.

MICROMETERS — Pneumatic Gage Co., Shoreham Bldg., 15th & H Sts., N. W., Washington, D. C. Booklet, 22 pages, 7x9½ in. Detailed information on the Solex pneumatic micrometer, capable of registering accurate measurements with or without mechanical contact.

MOLDING MATERIAL—Haveg Corp., Newark, Del. General Bulletin O-1, 12 pages, 8½x11 in. Covering the applications of Haveg molded construction material for corrosion-resistant chemical equipment.

PILLOW BLOCKS—Palmer-Bee Co., Detroit, Mich. Folder 7¼x10¼ in. Describing their new cast steel, self-aligning pillow blocks for steel, cement and paper mill service and for use on oil well equipment.

PLASTICS—Bakelite Corp., 247 Park Ave., New York, N. Y. "The Versatile Service of Bakelite Plastics," 16 pages, 3½x6¼ in. A concise review of the history of modern plastics from the time of their discovery up to the present day, telling what they are and how they are used.

PULLEYS—The Congress Tool & Die Co., Inc., 9030 Lumpkin Ave., Detroit, Mich. Folder, 8½x11 in. Engineering data and list prices on their complete line of F.H.P. pulleys and flexible couplings.

PYROMETERS—The Bristol Co., Waterbury, Conn. Bulletin 488, 12 pages. Describing the complete line of Bristol millivoltmeter pyrometers giving details of construction and operation.

RELAYS—Ward Leonard Electric Co., Mt. Vernon, N. Y. Bulletin 106. Data on their relays for two-wire and three-wire instrument control and their new heavy-duty midget relay. Also gives dimensions of new enclosures for these midget type relays.

ROLLER CHAINS—Link-Belt Co., 519 N. Holmes Ave., Indianapolis, Ind. Data Book No. 1757, 174 pages, 8½x11 in. Covering their Silverlink roller chains and sprockets for drives and conveyor uses. Practical information, application pictures and engineering data, including many new chains and features not previously published, are included.

MONEL—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Booklet, 8 pages, 8½x11 in. A well-illustrated booklet, "New Life for Old Equipment with Monel Gaskets, Pump Parts and Valve Trim," describing numerous typical examples of service records.

SPEED REDUCERS—Universal Gear Corp., 19th & Martindale Ave., Indianapolis, Ind. Catalog 40, 16 pages, 8½x11 in. Describing the heliocentric principle for speed reduction with rating tables on the various types of straight-line and right-angle drive speed reducers, together with tables of mounting dimensions.

STEEL—Inland Steel Co., 38 S. Dearborn St., Chicago, Ill., Bulletin 50, 4 pages, 8½x11 in. Data and information on Inland Ledloy, a new lead-bearing, open-hearth free-cutting steel.

STEEL SPECIFICATIONS—The International Nickel Co., Inc., 67 Wall St., New York, N. Y. Data Sheet Section VII, No. 1, "Society of Automotive Engineers Standard Specifications for Steels," to replace original information on this subject in their Data Book "Nickel Alloy Steels." Incorporates the latest additions and revisions in the S.A.E. steel specifications, plus some additional data on standard test specimens.

STEELS—Carnegie-Illinois Steel Corp., 434 Fifth Ave., Pittsburgh, Pa. Spiral bound booklet, "The Fabrication of USS Stainless Steels, Part I," 64 pages, 8½x11 in. Describes in detail the welding, riveting and soldering techniques applicable to all the various grades of USS stainless steels. Includes many striking illustrations. A section is also devoted to the design of joints.

STIFFNESS GAGE—Taber Instrument Co., North Tonawanda, N. Y. Bulletin 3802, 6 pages, 7¾x10½ in. Giving complete details of construction and operation of the Taber V-5 stiffness gage for paper, light metallic sheet and wire.

VALVES—Crane Co., 836 S. Michigan Ave., Chicago, Ill. Booklet, 8 pages, 8x9 in. Presenting "The Inside Story of Crane Plug Disc, Globe and Angle Valves."

VALVES—Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport, Conn. Bulletin 5-7000, 16 pages, 8½x11 in. Complete data on the Hancock redesigned line of forged and cast steel valves.

VARIABLE REDUCER TRANSMISSIONS — Stephens-Adamson Mfg. Co., Aurora, Ill. Bulletin No. 6838, 12 pages, 8½x11 in. Describing the motor speed-reducer units manufactured by this company with tables of ratings, dimensions and typical applications.

V-BELTS—The Manhattan Rubber Mfg. Div., Raybestos-Manhattan, Inc., Passaic, N. J. Bulletin 6868, 4 pages. Technical details of the construction and functions of Condor Whipcord V-belts. Also includes installation photographs, list prices, and a V-belt comparison table.

WELDING TIPS—P. R. Mallory & Co., 3031 E. Washington St., Indianapolis, Ind. Catalog describing their new type pointed welding tips and recently standardized line of spot welding tips and water-cooled holders.

STAINLESS STEEL—The American Rolling Mill Co., Middletown, Ohio. "Dyestuffs and Stainless are Friendly Enemies," describing the use of Armco stainless steels in textile equipment. General recommendations of stainless for various dye solutions are listed.

Books and Bulletins

Aluminum

DOUGLAS B. HOBBS. *295 pages, 6x9 in. Clothboard covers. Published by The Bruce Publishing Co., 524-544 N. Milwaukee St., Milwaukee, Wis. Price $3.00.*

Part I of this book is devoted to the history and manufacture of aluminum, the casting of aluminum alloys, the manufacture of basic wrought commodities, physical and chemical properties, and manufacturing operations such as forming, machining, joining and finishing.

Part II presents 25 "projects" wherein are set forth the materials required and detailed procedure for making various articles out of aluminum. Beginning with simple pieces such as house numbers and ash trays the parts become more complex including such pieces as door hardware, desk sets and coffee tables. Detailed construction directions are given, together with both line drawings and photographs of the finished piece. As explained by the author in his preface, the primary purpose of the book is to guide students in manual training courses and make them acquainted with the various methods of working aluminum. All of the projects were selected so that they could readily be made in a school or home workshop.

General Discussion on Lubrication and Lubricants

American Edition. Vols. 1-2 in one book. 1,200 pages, 6x9 in. 460 illustrations and graphs, 135 tables. Red clothboard covers. Published in the U. S. by the American Society of Mechanical Engineers, 29 West 39th St., New York, N. Y., with the permission of The Institution of Mechanical Engineers. Price $6.50.

This book reports the papers and discussions on lubrication and lubricants presented at the meeting sponsored by The Institution of Mechanical Engineers with the cooperation of 51 engineering societies, held at the Central Hall, Westminster, England, from October 13 to 15, 1937. The papers presented reviewed the present state of the science and practice of lubrication in order to correlate theory and practice, covered methods of bearing design, gave current views upon bearing metals, and discussed the significance of laboratory tests, including wear and friction tests.

In addition to the papers contributed by 163 leading authorities of 12 different countries, the book contains critical summaries of the papers by eminent engineers, gives a complete report of the discussions and lists over 400 bibliographic references to sources of more detailed treatment of special phases of the subject.

The book contains a considerable amount of illustrative material and much information which appears in print for for the first time. It is a collection of information which should prove of value to engineers interested in this subject.

Statistical Year Book International Tin Research and Development Council

1938 Edition. 200 pages, 8½x11 in. Blue clothboard covers. International Tin Research and Development Council, 149 Broadway, New York, N. Y. Price $1.50.

A compilation of all the available statistical information relating to the quantities of tin produced at mine or smelter and consumed in the various countries, and also showing as far as possible how much tin is used in the different industries, including tinplate and canning, solder, bearing metals, bronze, collapsible tubes, tinfoil and pewter. The data presented are for the most part annual statistics.

Among the new features in this edition is a chart arranged in the style of a genealogical table, by means of which tin may be traced from the form in which it leaves the smelter through all the processes to which it is subjected, to the numerous applications in which it is finally used.

The book is illustrated by many graphs, one of which represents the annual average price of tin over a period of about 70 years.

Elements of Diesel Engineering

ORVILLE ADAMS. *478 pages, 6x9 in. 332 illustrations. Clothboard covers. Published by The Norman W. Henley Publishing Co., 2 West 45th St., New York, N. Y. Price $4.00.*

This book covers modern high-speed Diesel engine construction, light-weight Diesel aircraft engines and Diesels of the automotive type. The author has avoided the mathematics of the Diesel cycle and confines himself to the practical operation of the engines. In successive chapters he describes and explains the operation of the various parts of the engine such as fuel injection nozzles, combustion chambers, injection pumps, ignition, pistons, cylinders and other parts. Line drawings and photographs illustrate the various constructions in use. Subsequent chapters go into the detail of the operation and maintenance of the various types of Diesel engines.

Grinding Practice

FRED H. COLVIN AND FRANK A. STANLEY. *310 pages, 6x9 in. 201 illustrations. Blue clothboard covers. Published by the McGraw-Hill Book Co., Inc., 330 West 42d St., New York, N. Y. Price $3.00.*

The authors have presented in this volume many recent developments that have taken place in grinding techniques and methods. Much space is given to the constructional and mechanical features of grinding machinery of all types.

While the book is intended to be a practical manual for the shop executive and planner, it can also help designing engineers gain some knowledge of the precision and quality of surface that can be expected in parts that are finished by grinding. Engineers who assign limits of accuracies of machine parts will find this book of much value.

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Heat Transfer of Porcelain Enamel Metals and Other Materials

J. C. BELL. *Bulletin No. 1, 28 pages, 6x9 in. Published by the Technical Staff of The Ferro Enamel Corp., Cleveland, Ohio.*

This technical bulletin on the subject of the heat transfer of porcelain enamel metals and other materials is a report of an original investigation conducted by the Research Laboratories of the Ferro Enamel Corporation. It is the first of a series of such bulletins that will appear as other investigations are completed. No regular publication dates will be followed, but the bulletins will be numbered consecutively for convenience of reference and filing.

Tin Researches

Recent reports published by the INTERNATIONAL TIN RESEARCH AND DEVELOPMENT COUNCIL. *Copies are available free from L. J. Tavener, 149 Broadway, New York, N. Y.*

"The Hot-Tinning of Fabricated Articles," by E. J. Daniels. Series B, No. 7, 12 pages.

"Discoloration and Corrosion in Canned Cream—Part II," by C. J. Jackson, G. R. Howat, and T. P. Hoar. Series A, No. 72. 7 pages.

"The Coating of Metals with Tin from the Vapour Phase (Stannising)," by B. W. Gonser and E. E. Slowter. Series A, No. 76, 17 pages.

Formulas for Elastic Bodies

Equations for Calculating Stress and Strain Due to Pressure

RAYMOND J. ROARK
Professor of Mechanics, University of Wisconsin

STRESSES due to pressure between elastic bodies are important in connection with the design or investigation of trunnions, expansion rollers for bridges, track rails, and such other parts as ball and roller bearings. For bearings of conventional design, the allowable loads are, of course, known from experience and service tests, and are given by the manufacturers. However, for conditions that represent a departure from common practice, whether in respect to size, material, or arrangement of parts, it is desirable to have some valid method of stress calculation on which to base estimates of safe loads.

The mathematical theory for the surface stresses and deformations produced by pressure between curved bodies was developed by Hertz (*Gesammelte Werke*, Vol. I, Leipzig, 1895), and the results of his analysis are supported by experiment. Formulas based on this theory give the maximum compressive stresses, which occur at the center of the common surface of contact, but not the maximum shear stresses, which occur in the interiors of the compressed bodies, or the maximum tensile stresses, which occur at the boundary of the contact-area and act perpendicularly thereto.

Both surface compression and internal shear were mathematically investigated, and the calculated value of the latter was experimentally checked by Thomas and Hoersch (See "Stresses Due to the Pressure of One Elastic Solid Upon Another," *University of Illinois Experiment Station Bulletin No. 212*, 1930). Formulas for the tensile and shear stresses for several cases, as well as formulas for compressive stress, dimensions of contact-area, and deformation, are given by Timoshenko ("Theory of Elasticity," *Engineering Societies Monographs*, McGraw-Hill, 1934).

The accompanying table of equations is a summary of formulas for the elastic stress and deformation produced under various conditions of loading by pressure between spheres, cylinders, and flat plates, and for the dimensions of the contact-area formed by the compressed surfaces. Except as otherwise noted, these equations are based on the theory of Hertz. In deriving them, it is assumed that the length of the cylinder and the dimension of a plate are infinite, and that the dimensions of the contact-area are very small compared with the dimensions of the bodies. For very short cylinders, narrow or thin plates, and materials like rubber that can sustain excessive deformation, the formulas may give erroneous results.

In order to make use of the formulas for purposes of design or investigation, it is necessary to have some knowledge of the relationship between the strength properties of the material and the calculated maximum stress at which failure may be considered to have occurred.

Many tests have been made to determine the bearing strength of balls and rollers, (See "Friction and Carrying Capacity of Ball and Roller Bearings." H. L. Whittemore and S. N. Petrenko. *Technologic Paper of the Bureau of Standards No. 201*, 1921; "The Bearing Value of Rollers," W. M. Wilson, *University of Illinois Experimental Station Bulletin No. 263*, 1934; and "Ball Bearings for Various Loads," Stribeck. *A.S.M.E. Transactions*, Vol. XXIX. 1907). However, there is difficulty in interpreting the results of such experiments because of the uncertainty as to just what constitutes failure of a specimen.

The tests show that even for very small loads there is some permanent deformation produced. This deformation increases progressively with increasing pressure, but there does not appear to be any point at which a marked acceleration in the rate of such increase occurs, as would be indicated by a sharp break in the load-set curve.

It is necessary, therefore, to select some arbitrary criterion for failure, despite the fact that there is no such criterion in general acceptance. In the experiments described by Wilson, "appreciable flow" was assumed to have occurred when the permanent set or "spread" amounted to 0.001 in. per in. in tests made by repeated rolling; whereas in some of the static tests a total permanent set of 0.00006 in. was taken to represent excessive deformation and, in other static tests, a graphical criterion was adopted that in one instance represented a permanent set of 0.0002 in. per in. In tests on ball bear-

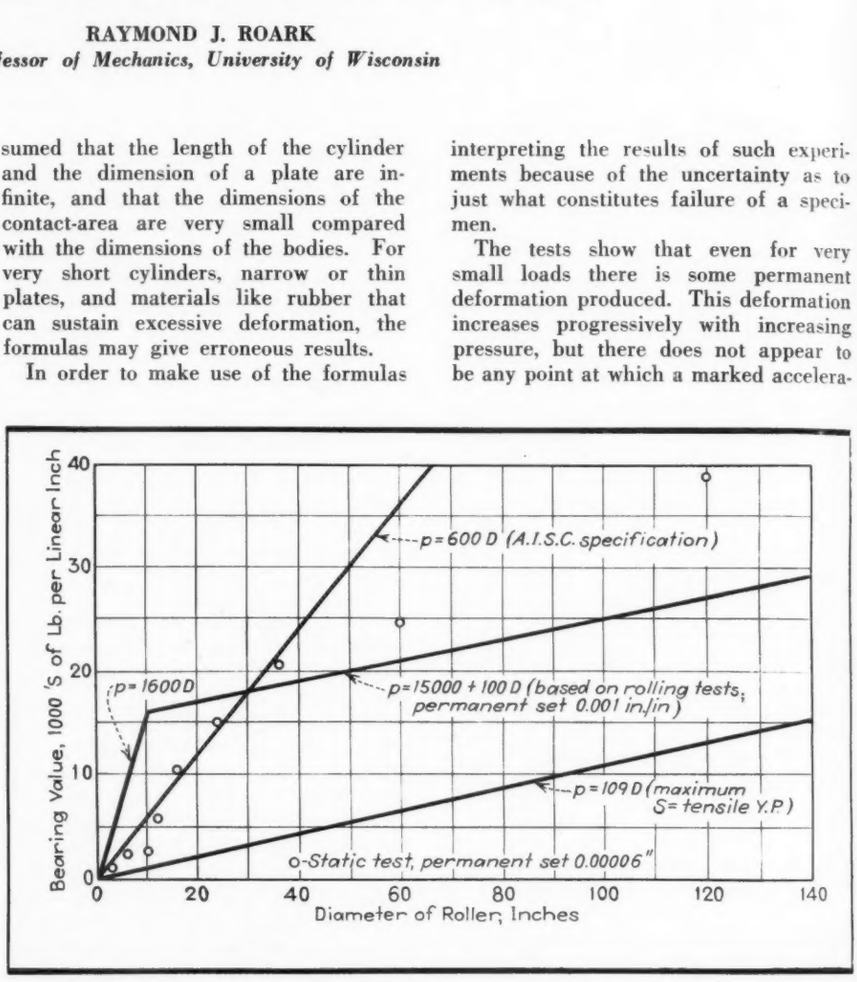


Fig. 1—Relation between diameter and bearing value for cylindrical steel roller on flat steel plate. Tensile yield point of material 33,800 lb. per sq.in.

ings. Whittemore and Petrenko took 0.0001 in. as the maximum permissible permanent set for the races.

It is obvious that the circumstances of use determine the degree of permanent deformation that would render a part unfit for use. Hence consideration of the possibility of fatigue failure, as in the instance of rail stresses, might dictate a much smaller allowable stress than would be permissible in, say, a bridge roller, in which the load is less variable and in which failure would merely cause ineffective functioning.

In general it seems clear that, because the stresses under consideration are at once tri-axial and highly localized, they may often greatly exceed the proportional limit and even the tensile yield point without causing excessive permanent deformation. This appears to be especially true of small bodies; as the scale increases, the unit stress necessary to produce a given plastic unit-deformation appears to decrease.

The accompanynig chart shows: (1) the results of static tests made by Henefine (reported in Wilson's "The Bearing Value of Rollers"); (2) the graph of the A.I.S.C. formula for the allowable bearing load on expansion rollers, $p = 600\ D$; (3) the graph of the equation $p = 109D$, obtained from the theoretical formula for Case 4 in the accompanying table of equations by taking $s_c = 33{,}800$ lb. per sq.in. (the tensile yield point of the steel in the rollers), $E = 30{,}000{,}000$ lb. per sq.in., and $v = 0.3$; (4) the composite graph of the equations $p = 1{,}600\ D$ and $p = 15{,}000 + 100\ D$, which Wilson found to represent the allowable load as determined by the repeated-rolling test.

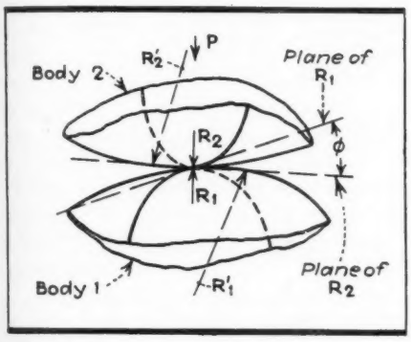


Fig. 2—General case of two bodies in contact. P equals total pressure

The wide disparities indicated by this chart represent differences in the criteria adopted for failure (i.e., excessive plastic yielding) rather than disagreement between theory and experiment.

In the table of equations, s_c = unit compressive stress; s_s = unit shear stress; s_t = unit tensile stress; a = radius of circular contact area for Cases 1, 2, and 3; b = width of rectangular contact area for Cases 4, 5, and 6; c = major semi-axis and d = minor semi-axis of elliptical contact area for Case 7 and for the general case of two bodies in contact; y = combined deformation of both bodies at each contact, along axis of load; v = Poisson's ratio, and E = modulus of elasticity. All dimensions are in in., and all forces in lb. Subscripts 1 and 2 refer to Bodies 1 and 2 respectively.

At the point of contact between two bodies, as represented in the accompanying sketch, the minimum and maximum radii of curvature are R_1 and R'_1 for Body 1, and R_2 and R'_2 for Body 2. Then $1/R_1$ and $1/R'_1$ are principal curvatures of Body 1, and $1/R_2$ and $1/R'_2$ are principal curvatures of Body 2, in each body the principal curvatures being mutually perpendicular. The plane containing curvature $1/R_1$ in Body 1 make the angle γ with the plane containing curvature $1/R_2$ in Body 2. P = the total pressure. Then:

$$\text{Max. } s_c = \frac{1.5P}{\pi cd},\quad c = \alpha\sqrt[3]{\frac{P\delta}{k}}$$

$$d = \beta\sqrt[3]{\frac{P\delta}{k}},\quad y = \lambda\sqrt[3]{\frac{P^2}{k^2\delta}}$$

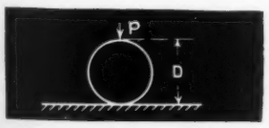
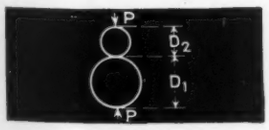
$$\text{where } \delta = \frac{4}{\frac{1}{R_1}+\frac{1}{R_2}+\frac{1}{R'_1}+\frac{1}{R'_2}}$$

$$k = \frac{2}{3}\left[\frac{E_1}{1-v_1^2}+\frac{E_2}{1-v_2^2}\right]$$

The values of α, β, and λ are obtained from the accompanying table, in which: $\theta = \text{arc cos } \frac{1}{4}\ \delta\sqrt{m^2+n^2+2mn\cos 2\phi}$

Where $m = 1/R_1 - 1/R'_1$
and $n = 1/R_2 - 1/R'_2$

θ	α	β	λ
30	2.731	0.493	1.453
34	2.397	0.530	1.550
40	2.136	0.567	1.637
45	1.926	0.604	1.709
50	1.754	0.641	1.772
55	1.611	0.678	1.828
60	1.486	0.717	1.875
65	1.378	0.759	1.912
70	1.284	0.802	1.944
75	1.202	0.846	1.967
80	1.128	0.893	1.985
85	1.061	0.944	1.996
90	1.000	1.000	2.000

CONDITIONS	FORMULAS FOR DIMENSION OF CONTACT AREA AND MAXIMUM STRESS
 Sphere on a flat plate. P = total load.	$a = 0.721\sqrt[3]{PD\left[\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right]}$; Max. $S_c = 0.918\sqrt[3]{\frac{P}{D^2\left[\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right]^2}}$; If $E_1 = E_2 = E$ and $v_1 = v_2 = 0.3$, $a = 0.881\sqrt[3]{\frac{PD}{E}}$, Maximum $S_c = 0.616\sqrt[3]{\frac{PE^2}{D^2}}$, Maximum $S_t = 0.133$ Maximum S_c, $y = 1.55\sqrt[3]{\frac{P^2}{E^2D}}$ (Ref. 1) Maximum $S_s = \frac{1}{3}$ Max. S_c, at depth $\frac{1}{4}\ D$ below surface of plate. (Approximate values, from Ref. 1 and 2)
 Sphere on a sphere. P = total load.	$a = 0.721\sqrt[3]{P\left(\frac{D_1D_2}{D_1+D_2}\right)\left[\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right]}$, Maximum $s_c = 0.918\sqrt[3]{P\frac{\left(\frac{D_1+D_2}{D_1D_2}\right)^2}{\left[\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right]^2}}$; $y = 1.04\sqrt[3]{\frac{P^2(D_1+D_2)}{D_1D_2}\left(\frac{1-v_1^2}{E_1}+\frac{1-v_2^2}{E_2}\right)^2}$ If $E_1 = E_2 = E$ and $v_1 = v_2 = 0.3$, $a = 0.881\sqrt[3]{\frac{P}{E}\frac{D_1D_2}{D_1+D_2}}$, Maximum $S_c = 0.616\sqrt[3]{PE^2\left(\frac{D_1+D_2}{D_1D_2}\right)^2}$, Maximum $S_s = \frac{1}{3}$ maximum S_c, Maximum $S_t = 0.133$ maximum S_c, $y = 1.55\sqrt[3]{\frac{P^2}{E^2}\frac{(D_1+D_2)}{D_1D_2}}$

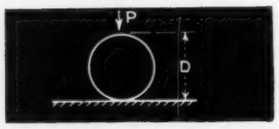
CONDITIONS	FORMULAS FOR DIMENSION OF CONTACT AREA AND MAXIMUM STRESS

Sphere in spherical socket.
P = total load.

$$a = 0.721\sqrt[3]{P\,\frac{D_1D_2}{D_1 - D_2}\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]} \qquad \text{Maximum } s_c = 0.918\sqrt[3]{P\,\frac{\left(\frac{D_1 - D_2}{D_1D_2}\right)^2}{\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]^2}}$$

If $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = 0.3$, $a = 0.881\sqrt[3]{\frac{P}{E}\,\frac{D_1D_2}{D_1 - D_2}}$ Maximum $S_c = 0.616\sqrt[3]{PE^2\left(\frac{D_1 - D_2}{D_1D_2}\right)^2}$,

Maximum $S_s = \frac{1}{3}$ maximum S_c, Maximum $S_t = 0.133$ maximum S_c, $y = 1.55\sqrt[3]{\frac{P^2}{E_2}\,\frac{(D_1 - D_2)}{D_1D_2}}$



Cylinder on flat plate.
p = load per linear inch.

$$b = 1.6\sqrt{pD\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]} \qquad \text{Maximum } s_c = 0.798\sqrt{\frac{P}{D\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}}$$

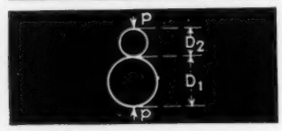
Total compression of cylinder between two plates is:

$$\delta D = 4p\left(\frac{1-\nu^2}{\pi E}\right)\left(\frac{1}{3} + \log_e\frac{2D}{b}\right) \qquad \text{(Ref. 3)}$$

If $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = 0.3$, $b = 2.15\sqrt{\frac{pD}{E}}$, Maximum $S_c = 0.591\sqrt{\frac{pE}{D}}$

For $E = 30{,}000{,}000$, $\nu_1 = \nu_2 = 0.25$, $b = 0.0004\sqrt{pD}$ Maximum $s_c = 3190\sqrt{\frac{p}{D}}$,

Maximum $s_s = 958\sqrt{\frac{p}{D}}$ at depth $0.393b$ below surface of plane. (Approximate formula from Ref. 4)



Cylinder on cylinder.
Axes parallel.
p = load per linear inch.

$$b = 1.6\sqrt{p\,\frac{D_1D_2}{D_1 + D_2}\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]} \qquad \text{Maximum } s_c = 0.798\sqrt{\frac{p\,\frac{D_1 + D_2}{D_1D_2}}{\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}}$$

If $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = 0.3$, $b = 2.15\sqrt{\frac{p}{E}\,\frac{D_1D_2}{D_1 + D_2}}$ Maximum $s_c = 0.591\sqrt{pE\,\frac{D_1 + D_2}{D_1D_2}}$

Cylinder in circular groove.
p = load per linear inch.

$$b = 1.6\sqrt{p\,\frac{D_1D_2}{D_1 - D_2}\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]} \qquad \text{Maximum } s_c = 0.798\sqrt{\frac{p\,\frac{D_1 - D_2}{D_1D_2}}{\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}}$$

If $E_1 = E_2 = E$ and $\nu_1 = \nu_2 = 0.3$, $b = 2.15\sqrt{\frac{P}{E}\,\frac{D_1D_2}{D_1 - D_2}}$, Maximum $s_c = 0.591\sqrt{pE\,\frac{D_1 - D_2}{D_1D_2}}$

Cylinder on cylinder.
Axes at right angles.
P = total load.

$$c = \alpha\sqrt[3]{P\,\frac{D_1D_2}{D_1 + D_2}\left[\frac{1-\nu_1^2}{E_1} + \frac{1-\nu_2^2}{E_2}\right]}, \quad d = \beta c, \qquad \text{Maximum } S_c = \frac{1.5\,P}{\pi cd}$$

$$y = \lambda\sqrt[3]{\frac{P^2}{\left(\frac{E_1}{1-\nu_1^2} + \frac{E_2}{1-\nu_2^2}\right)^2}\,\frac{(D_1 + D_2)}{D_1D_2}}$$

where α and β and λ depend on ratio D_1/D_2 and have values as follows:

$D_1/D_2 =$	1	1½	2	3	4	6	10
$\alpha =$	0.908	1.045	1.158	1.350	1.505	1.767	2.175
$\beta =$	1	0.765	0.632	0.482	0.400	0.308	0.221
$\lambda =$	2.080	2.060	2.025	1.950	1.875	1.770	1.613

If $E_1 = E_2 = 30{,}000{,}000$, $\nu_1 = \nu_2 = 0.25$, $c = 0.00397\,\alpha\sqrt[3]{P\,\frac{D_1D_2}{D_1 + D_2}}$

For these values of E and ν and for values of D_1/D_2 between 1 and 8,

$$\text{Maximum } s_s = \frac{11{,}750}{\left(\frac{R_1}{R_2}\right)^{0.271}}\sqrt[3]{\frac{P}{R_2^2}} \quad \text{where } R_1 = \frac{1}{2}\,D_1,\ R_2 = \frac{1}{2}\,D_2$$

(Approximate formula, from Ref. 4)

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